MACHINE DRAWING

BOOK I.

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£18

WORKINGTON COUNTY TECHNICAL & SECONDARY SCHOOL

FLOREAT CASTELLUM ET OPPIDUM

Form IaT

PRIZE awarded to

William Bowe Dobie

for English and Drawing

A. B. Cole, M.A.

Principal

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—Rivets and Riveted Joints.
—Rolled Steel Sections and their Applications.
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INTRODUCTION.

A MACHINE DRAWING to have any value must be accurate, and since accuracy cannot be obtained without neatness, it follows that to be neat and accurate should be the aim of the student. He should also practise drawing to scale from the commencement, since it is as easy to draw to scale as to make a drawing full size. The scales fixed for the various drawings are so arranged, that the student can get the complete drawing, with the additional views required, on a sheet of paper of half imperial size, 22″ × 15″.

Most of the drawings given are *shadow lined*, which gives a very effective appearance to them. The student should not attempt to do this until he is able to make an accurate drawing, with neat lines of equal thickness throughout. The method of shadow lining is explained below.

Section lines are used to show any piece cut by a section plane; these lines must always be drawn with the 45° set square. Where two pieces of material are in contact and in section, the section lines must be drawn in opposite directions.

No special method of section lining has been adopted except in the case of brass, which is used so often in connection with cast-iron. The lines in this case are alternately full and dotted. Similar lines but of coarser pitch have been used for stone.

Wrought-iron and cast-iron sections are shown in the same manner by full lines. The attempt to represent each material by a different method of section lining, leads to confusion, and spoils the appearance of the drawing. In a drawing for use in the workshop, the material of which the various parts are to be made must be distinctly *written down*. No fanciful method of section lining would be accepted.

The best way of representing the various materials is by colouring. In all cases the outside views, plans and elevations should have only a light wash of colour, and the sectioned parts should be much darker. The centre lines of finished drawings are usually drawn in crimson lake, and the dimension lines in prussian blue; but some engineers use blue centre lines for revolving parts, such as shafts.

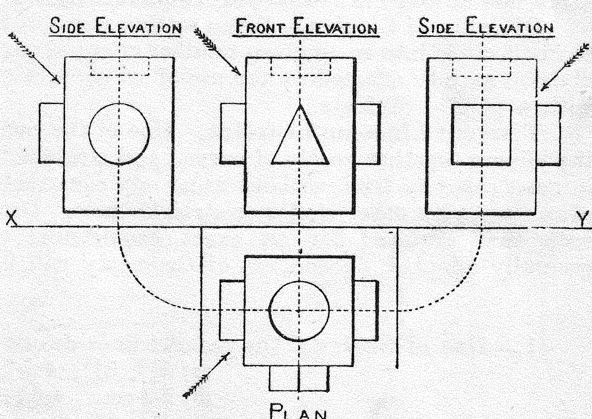
The following colours are almost universally used:—(see Plate LXIX.).

Cast-iron and lead (elevations, &c.)	Payne's Grey	Brass	Indian Yellow
,, ,, (sections)	Indigo	Copper	Crimson Lake tinged with Gamboge
Wrought-iron (elevations, &c.)	Prussian Blue	Wood	Burnt Sienna
,, (sections)	,, ,,	Brickwork	Crimson Lake
Steel	Purple, made of Prussian Blue tinged with Crimson Lake	Stone	Light tint of Gamboge

It is useless to attempt here to teach the student to colour, or even to make a machine drawing; he is advised to join one of the many excellent classes established throughout the country, where he can have the benefit of the advice and assistance of an experienced teacher. A few hints are given in the notes to Plates II., LXVIII., and LXIX.

SHADOW LINING.

The method of shadow lining a drawing is shown in the accompanying sketch. The example selected is a square prism with projecting pieces on three of its vertical faces, and a circular recess in the top face. The object is supposed to be illuminated by parallel rays of light, the directions of which make in the elevation and plan respectively angles of 45° to the horizontal and vertical planes of projections, as shown. In the *front elevation* the right hand and bottom lines are thickened, except in the elevation of the cylindrical projecting piece; because in this case the bottom line is a *contour* of the cylinder and not the intersection of two surfaces. *The contours of curved surfaces should never be shadow lined.*



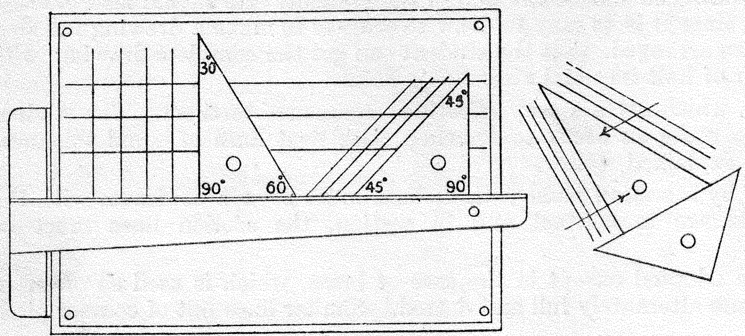
In the *plan* the top and right hand lines of the figure are thickened, except in the case of the cylindrical recess, where the shadow line is on the opposite side; the thick line gradually vanishing at the points where lines, parallel to the plan of the rays, are tangential to the circle.

The directions of the rays as shown in the side elevations are correct, because the vertical planes upon which they are projected are turned through an angle of 90° It is usual, however, to take the projections of the rays of light in the *same* direction in all the views, plans and elevations alike. Every view therefore is shadow lined, as in the front elevation given in the sketch. The appearance of the drawing is equally effective, and the making of it much simplified.* *Always place shadow lines so that their thickness is outside the outline of the drawing.*

* Where a drawing is shadow lined this method is adopted in the book.

Plate I.—PLANE GEOMETRY.

ALL the drawings should be made on sheets of paper of half imperial size, viz., $22'' \times 15''$. The materials required are, drawing board $23'' \times 16''$, T square with blade $24''$ long, set square with one angle $60°$, set square with two angles $45°$ (a useful size of set squares is $6''$ on the shortest side). A case of mathematical instruments is also required containing compasses capable of drawing circles from $\frac{1}{8}''$ to $9''$ radius, a pair of dividers, drawing pen, protractor, boxwood or cardboard scales, india rubber, drawing pins and H pencils.



In Machine Drawing it is not necessary to adopt geometrical constructions for the drawing of all lines and curves in their required relative positions; and in the accompanying illustration it is shown how straight lines which are parallel to or at right angles to other straight lines may be drawn mechanically by means of the T square and set squares.

Plain Scales.—An engineer's steel rule as used in the workshop differs widely from the draughtsman's scale. In the workshop only full size or actual dimensions are dealt with; but, since drawings of very large machines and engines are made on paper of very limited size, it is necessary to make them considerably less than real or full size without altering the *relative* sizes of the parts, and hence a draughtsman must be provided with a measuring appliance more general in its application than the ordinary inch rule used in the workshop.

The scale as used in a drawing office is 12 inches long, and made of boxwood or ivory, and on its two sides are marked all the scales required for general work; but, for several reasons, it is advisable that the student should use good clearly printed unvarnished cardboard scales—with scales drawn on one side only.

On a correctly constructed scale **only the first division** or interval of length **is sub-divided,** all the other equal divisions are left blank. The sub-divided length may be :—(*a*) one inch long, divided into $\frac{1}{16}$ths—in which case the scale is *full size*; or (*b*) some fraction of an inch which represents an inch, and is sub-divided into 8 or 16 equal parts; or (*c*) **some length such as $\frac{1}{4}''$, $\frac{1}{2}''$, $\frac{3}{4}''$, $1''$, $1\frac{1}{2}''$, $2''$, $3''$ which represents one foot,** and is sub-divided into parts representing inches and fractions of an inch. The zero of the scale is at the inner end of the sub-divided length, and not at the end as in the case of an ordinary rule. On one side of a drawing scale there are not more than four different scales, and of the two marked along the *same* edge one is usually double the other.

Figs. 1, 2, 3, 4 indicate the general arrangement of a scale and the method of obtaining any required dimension from any of the given scales. Dimensions must be transferred from the scale direct to the drawing paper by placing the edge of the scale on the line along which a definite length is required. The compasses should never be placed on the scale for the taking and transference of dimensions.

If lengths be used on a drawing which are taken from a scale, where **$1\frac{1}{2}'' =$ 1 foot**; then the drawing will be **$\frac{1}{8}$ full size** since a length of $1\frac{1}{2}''$ which on the drawing represents $12''$ is $\frac{1}{8}$ of 12 inches.

If **a** inches is the length on a scale which represents 12 inches, then the scale of the drawing expressed fractionally is $\frac{a}{12}$.

Simple Geometrical Figures.—It is very often necessary to draw regular figures such as the Square, Hexagon and Octagon mechanically without any construction, and the examples given in **Fig. 5** show that the required angles may be obtained from the set squares. Of course, if the set squares are not accurate the figures will be distorted and the inscribed and circumscribing circles drawn to test their accuracy will make the errors evident at once.

Note.—It should be remembered that in the drawing of any figure the **construction lines** must be **thin full lines** not dotted lines—and the **required figure** finished in **sharp bold lines.**

Division of Circles.—When setting out bolt holes on any flange or circular disc, it is necessary to divide first the bolt circle into some given number of equal parts, and very often the number of parts required is such that the division may be effected by the use of set squares (see Fig. 6). In other cases the division may be effected by trial with a pair of dividers.

Tangent Lines and Circles.—One of the early difficulties, experienced by the student in machine drawing, is the joining together neatly of curved and straight lines. Practice enables the experienced draughtsman to do this, in most cases by trial, without using any construction for finding the centres of the tangent circles or their points of contact with other circles or straight lines. Usually the time taken by the *trial* method is so small compared with that required for an exact geometrical construction, that in practical work the former method is generally adopted. (Students of Geometry will be acquainted with the ordinary geometrical constructions.)

EXERCISES.

1.—Use of Scales. On parallel lines drawn with the T square mark off the following lengths :—

(*a*) $5\frac{1}{2}''$, $3\frac{3}{4}''$, $4\frac{7}{16}''$, $2\frac{9}{32}''$. *Full size.*
(*b*) 9·3 cms., 4·65 cms., 87 mms. *Full size.*
(*c*) $2'-3''$, $3'-4\frac{1}{2}''$, $1'-5\frac{3}{4}''$. *Scale* $3'' = 1$ *foot.*
(*d*) $9\frac{3}{4}''$, $6\frac{1}{8}''$, $11\frac{5}{16}''$. *Scale* $\frac{3}{8}$ *full size.*

N.B.—Make cardboard drawing scales for all the above exercises excepting (*b*).

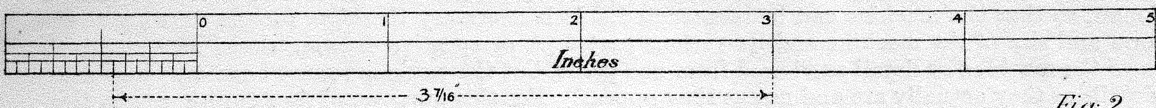
2.—Simple Geometrical Figures, Fig. 5. Draw accurately the four figures to the dimensions given. *Scale full size.*

3.—Division of Circles, Fig. 6. Draw five circles each $3''$ diameter and divide them into 3, 4, 6, 8, and 12 equal parts respectively by means of set squares, and divide three other circles into 5, 7, and 11 equal parts respectively by trial.

4.—Tangent Lines and Circles. Draw the Figs. 7, 8, 9, 10, *full size,* and show the required shapes in sharp bold lines.

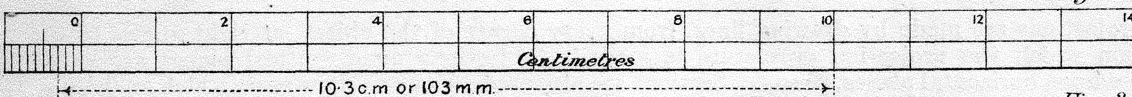
PLAIN SCALES.

Fig. 1.



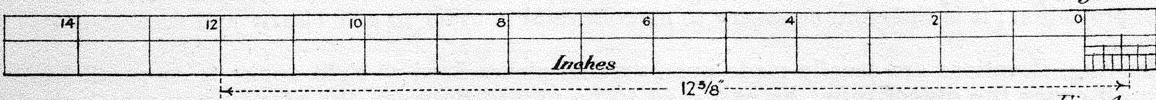
Full size.

Fig. 2.



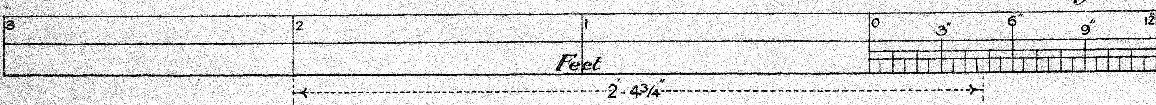
Metric Scale Full size.

Fig. 3.



3/8 full size.

Fig. 4.

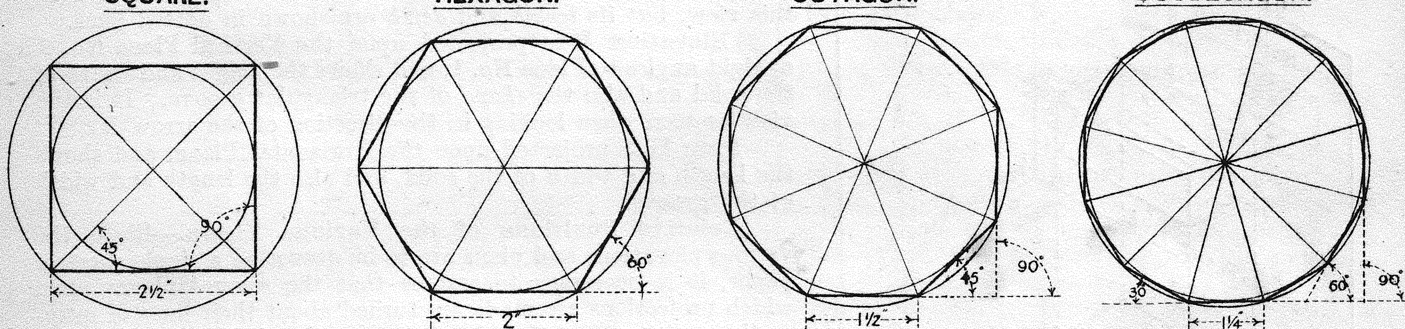


1½" = 1 Foot or 1/8 full size.

SIMPLE GEOMETRICAL FIGURES.

Fig. 5.

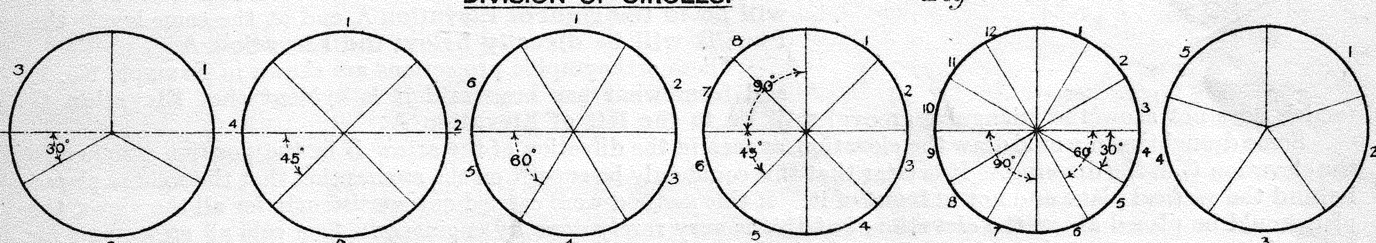
SQUARE. HEXAGON. OCTAGON. DUODECAGON.



The above figures will serve as exercises in neatness, and must be drawn by means of the T. and set-squares only; and the Circles afterwards drawn to test the accuracy of the drawings.

DIVISION OF CIRCLES.

Fig. 6.



TANGENT LINES AND CIRCLES.

DRAWN BY TRIAL.

Fig. 7.

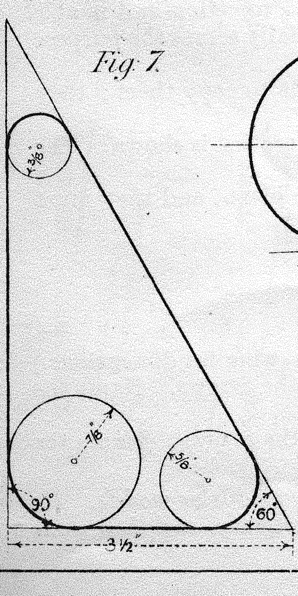


Fig. 8.

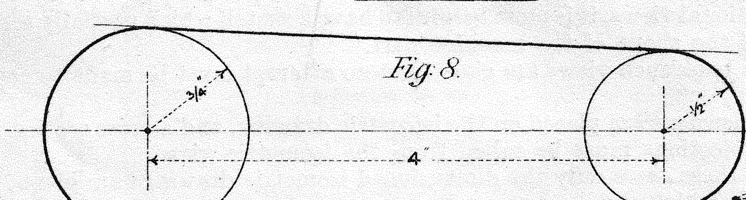


Fig. 9.

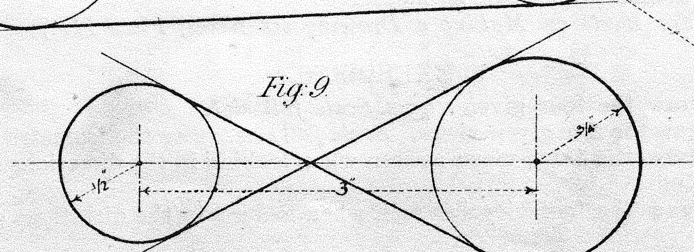
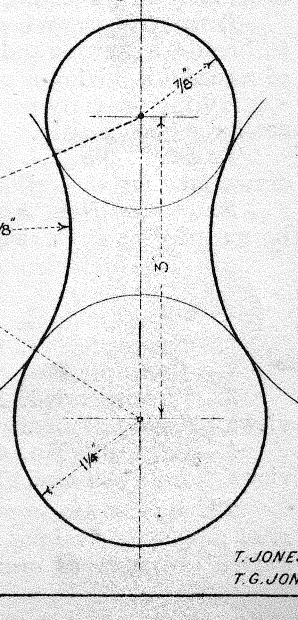


Fig. 10.



See opposite page for list of Exercises.

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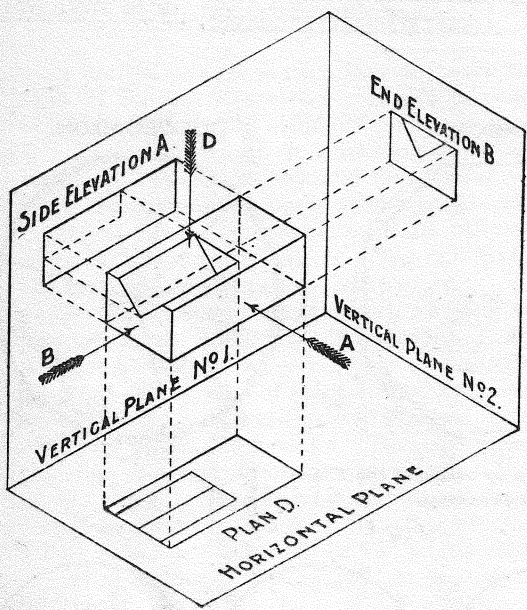
Plate II.—PRINCIPLES OF PROJECTION.

THE chief purpose of a machine drawing is to enable the inventor or designer of a machine to convey his ideas to the workman, so that the machine can be constructed. It is necessary therefore for the drawing to give the exact form and size of the machine and its various parts. A perspective drawing shows at a glance the general appearance of the machine or detail as viewed from one point, but the workman must have drawings which show the various details as they actually are and not as they appear. For this purpose, **orthographic projections** are used.

Orthographic projections are made by drawing lines, from every corner of the object, at right angles to the plane or surface upon which the projection or picture is made. Projections upon vertical planes are called **elevations,** and those upon horizontal planes, **plans.** Sections are also used, chiefly to show parts which are hollow. In the case of sections, the planes upon which the drawings are made pass through the solid. No solid can be fully represented by one projection ; it is therefore necessary to make several in order to give clearly and accurately its exact form and size.

In machine drawing, elevations are usually made upon vertical planes fixed at right angles to or parallel to each other, **but in all cases the object must be assumed to be placed between the plane upon which the elevation is drawn and the eye of the spectator.**



The accompanying isometric illustration is given to make clear the meanings of the words *projection*, *elevations* and *plans*.

Elevation A is projected upon the Vertical Plane No. 1 and shows the length and height of the solid. It is the view as seen when looking in the direction of the arrow A. The triangular groove in the top face of the solid cannot be seen in this view, but its *length* and *depth* are shown in dotted lines.

Elevation B is projected upon the Vertical Plane No. 2 at right angles to Plane No. 1, and shows the height and width of the solid and also the *shape* of the triangular groove. It is the view as seen when looking in the direction of the arrow B.

Plan D is projected upon the Horizontal Plane, and shows the length and width of the solid, and also the length and width of the groove.

Relative Positions of the Various Views.—Since the various elevations and plans are to be drawn on a single sheet of paper, it is necessary to assume that the several planes upon which projections are made are turned about their lines of intersections with one of the planes (assumed to be fixed) until they coincide with it. In the particular example considered the Vertical Plane No. 1 is assumed to be fixed, and the **Elevation B will be to the right of Elevation A** and at the same level ; the **Plan D will be directly below the Elevation A.**

These orthographic projections are shown in Example No. 1, and from what has been said it is evident that **Elevation C** (which does not show the triangular groove) **will be to the left of Elevation A.**

Some draughtsmen would draw the elevation as seen in the direction of the arrow B in the position assigned to the elevation C in the drawing. It is clear that this could only be correct on the assumption that the solid is placed **behind** the vertical plane and not in **front** of it. If this method were carried out consistently for all the views, the plan would be placed above the elevation, but this is very rarely done by engineers. As a rule all projections for industrial purposes are made on the understanding that the object projected is in the first pair of planes of projection, i.e., **above** the ground, and in **front** of the vertical plane. Serious errors occasionally occur through this want of uniformity in projecting elevations.

Isometric Drawings.—Until a student becomes quite familiar with the Principles of Projection and is able to " read " a drawing and make additional views, it is most helpful to have a sketch which partially serves the purpose of a model in giving a good idea of the shape of the machine part.

During the early portion of this book such views are given, but no attempt must be made to copy them ; they are for reference only.

Example No. 2.—Here the dimensions are placed on the isometric drawing, and a cross section is shown. The dimensions for the orthographic projections must be taken from the isometric view.

Examples Nos. 3 and 4.—In these cases only the dimensioned isometric drawings are given, and they must be regarded as equivalent to models which can be measured.

(*For Hints on Making a Drawing see Notes, Plate III.*)

EXERCISES.

1.—**Example No. 1.** Draw the four given views. *Scale full size.*

2.—**Example No. 2.** Draw the five given views. *Scale full size.* (See the isometric drawing for dimensions.)

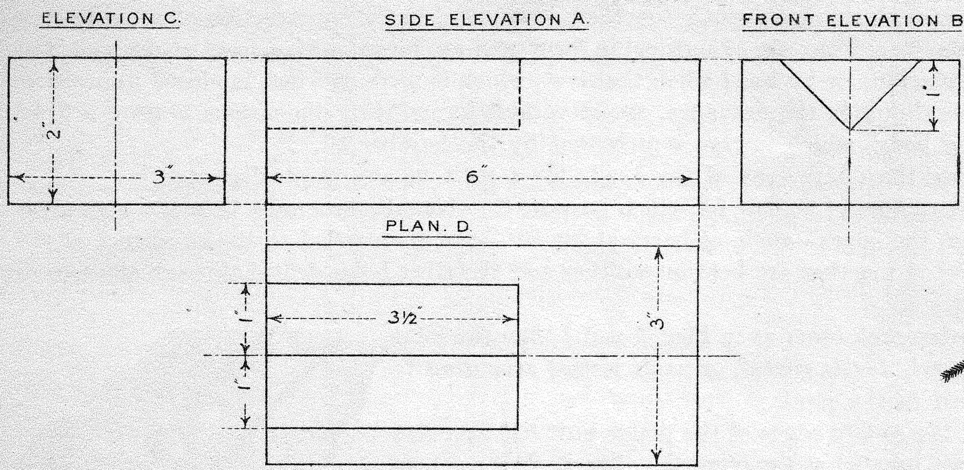
3.—**Example No. 3.** Draw the three views as seen when looking in the directions of the arrows. Name the views. *Scale full size.*

4.—**Example No. 4.** Draw the four views as seen when looking in the directions of the arrows. Name the views. *Scale full size.*

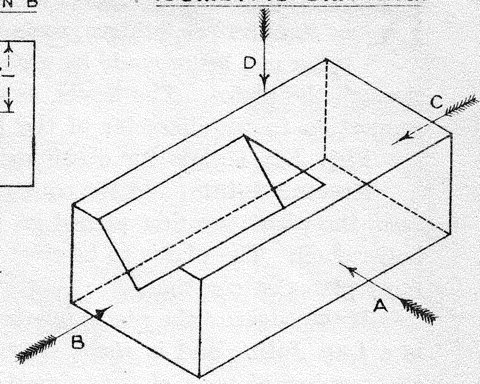
The student may supplement the work on this plate by direct reference to simple engineering details or models. He must first make freehand dimensioned sketches of the outside and sectional views considered necessary, and then make finished dimensioned drawings by reference to his own sketches.

PRINCIPLES OF PROJECTION.

EXAMPLE No. 1.

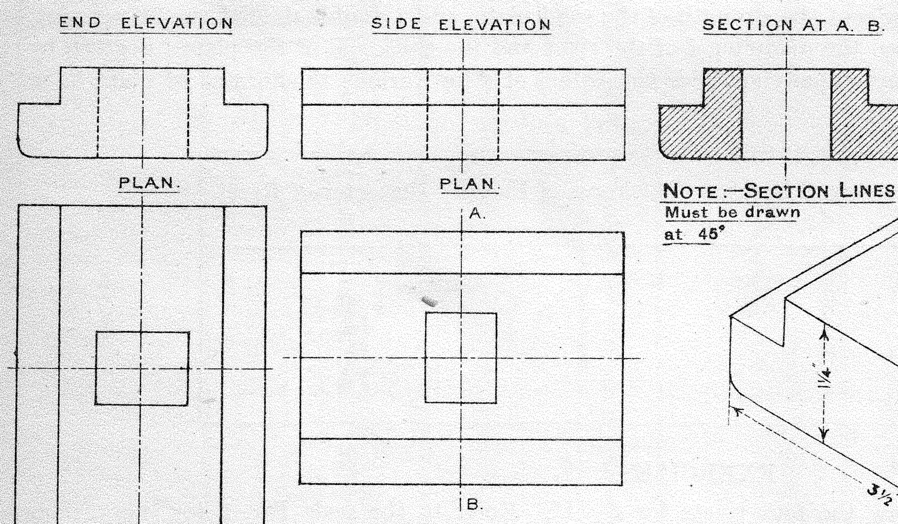


ISOMETRIC DRAWING.



The ISOMETRIC DRAWINGS simply serve the purpose of models and must not be drawn.

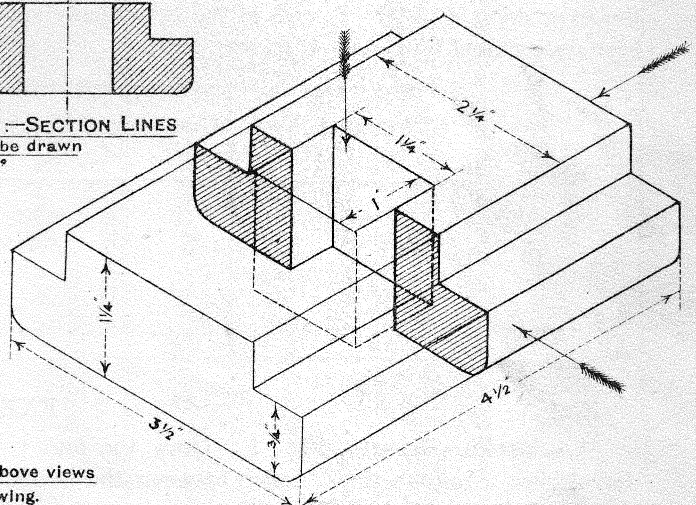
EXAMPLE No. 2.



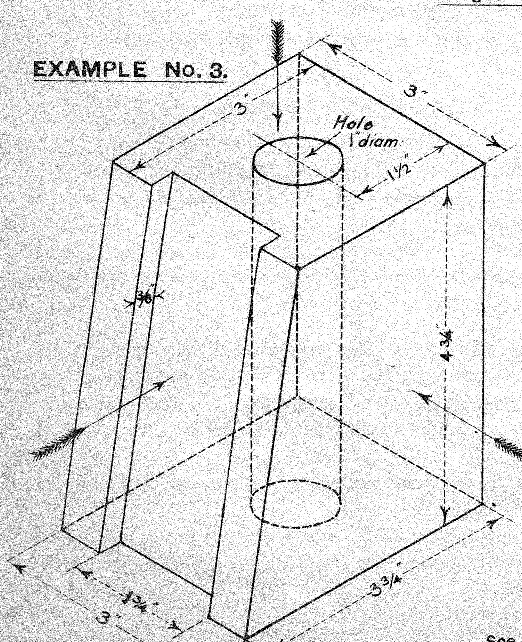
Note.—The dimensions for the above views are given on the Isometric Drawing.

ISOMETRIC DRAWING.

SHOWING SECTION.

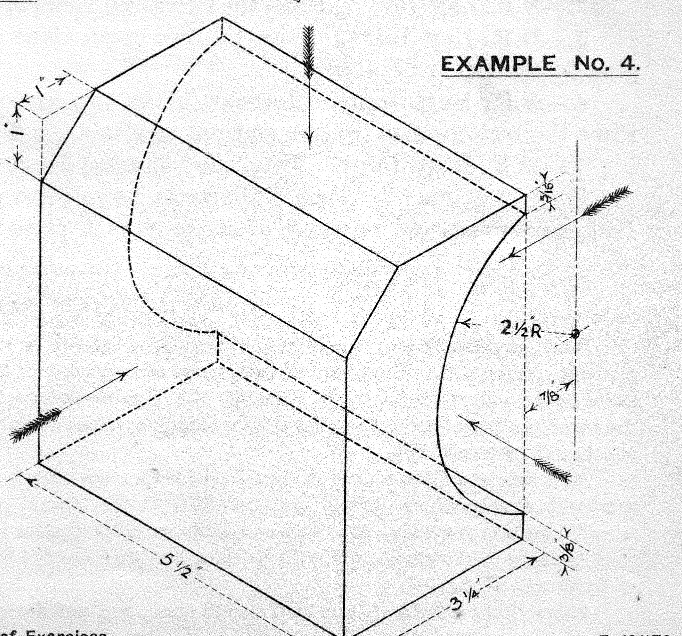


EXAMPLE No. 3.



Refer to these Isometric Drawings for dimensions when making the required orthographic projections.

EXAMPLE No. 4.



See opposite page for list of Exercises.

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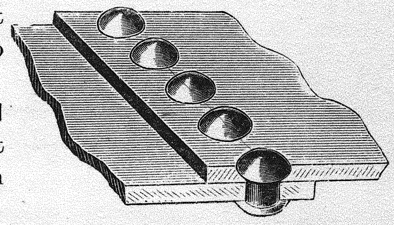
Plate III.—RIVETS AND RIVETED JOINTS.

RIVETS are used to connect metal plates permanently together, as required in the construction of wrought-iron or steel boilers, bridges, roofs, &c. They are cylindrical in form, and are forged with a head at one end, the other one being made by a machine or by hand when the rivet, which is made red-hot, is placed in position through the plates. The heads, as shown in the drawings, are of various forms, their dimensions bearing a fixed proportion to the diameter of the body, which is here represented by the number 10.

Fig. 2 is a scale for obtaining the dimensions of the heads for a rivet of any given diameter.

Riveted Joints.—In the construction of boilers, for which purpose the riveted joints must be made with great care, the plates are first planed on the edges—angle of bevel about 80°—and then rolled to the curvature of the boiler shell. The plates to be riveted together are held in position and the rivet holes drilled through the two or more plates at one time.



If the edges of the plates overlap each other as in Figs. 3 and 4 then the joint is a **Lap Joint,** and is *single riveted, double riveted,* or *treble riveted* according to the number of rows of rivets used in the joint.

In the case of a **Butt Joint,** the square edges of the plates butt together and the joint is covered by one or two parallel strips of plate. Fig. 5 shows a butt joint with two cover straps; and, since only one line of rivets passes through each plate, is a *single-riveted* joint.

A single-riveted joint is about 60 per cent the strength of the solid plate, and a double-riveted one 75 per cent.

For boiler joints the relation between the diameter d of the rivet and the thickness of the plate t is given by the expression $d = 1{\cdot}2\sqrt{t}$, and in the accompanying table the values of d for various thicknesses of plate have been determined by means of it.

Thickness of Plate. t	Diameter of Rivet. d	Thickness of Plate. t	Diameter of Rivet. d
$\frac{1}{4}''$	$\frac{9}{16}''$	$\frac{9}{16}''$	$\frac{7}{8}''$
$\frac{5}{16}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{15}{16}$
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{16}$
$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$1\frac{1}{8}$
$\frac{1}{2}$	$\frac{7}{8}$	1	$1\frac{3}{16}$

EXERCISES.

1.—**Various Rivets, Fig. 1.** Draw the four rivets for $d = 1''$. Refer to the scale Fig. 2 for the various dimensions. Assume the distance between the heads to be $1\frac{1}{4}''$.

2.—**S.R. Lap Joint.** Draw the two given views of Fig. 3, showing in the plan about five rivets. *Scale full size.*

3.—**D.R. Lap Joint.** Draw the two given views of Fig. 4, and add an edge elevation by projecting from the plan to the right. *Full size.*

4.—**S.R. Butt Joint.** For each of the two sectional elevations, Figs. 5 and 6, add the plan. *Scale full size.* Place the sections side by side and not as given.

5.—**D.R. Butt Joint.** From the following dimensions draw the sectional elevation and the plan of the joint. Thickness of plates $\frac{5}{8}''$; rivets $\frac{7}{8}''$ diameter; two cover plates, each $\frac{7}{16}''$ thick and $8\frac{1}{2}''$ wide; longitudinal pitch $2\frac{1}{2}''$; distance between the two lines of rivets in each plate $1\frac{1}{2}''$. *Scale half full size.*

HINTS ON MAKING A DRAWING.

Most machine parts are made symmetrical about a central line or axis. Generally the centre line is the first one made in commencing a drawing. When circles occur in any of the views they should be drawn first. The projections of these circular parts in the other views may be found by the T or set square, without measuring again from the scale or rule. No circle should be drawn until its centre has been fixed by making two lines at right angles to each other. Sometimes the first line made in the drawing is a base or ground line.

As a rule one view cannot be completed before the others are commenced; portions of each are made in turn, and the drawing gradually developed by passing from one view to the other.

No definite general instructions can be given in the matter; judgment must always be exercised, and experience is the best guide.

All lines in the drawing should be clear and thin until it is completed. The drawing may then be *lined* in, either in indian ink or by pencil, as desired.

Centre lines should always be thin full lines, and not dotted as shown in the drawings.

Plate IV.—EXAMPLES OF RIVETED WORK.

IN riveted structures it is often necessary to connect plates together in ways differing from those shown on Plate III. For instance, the plates may be parallel to each other, but at some given distance apart, or they may be at right angles, or any other angle, to each other. The connections are effected by the use of rolled bars of iron or steel of a section suited to the required purpose.

Rolled Steel Sections.—Examples of the more important rolled sections are given in Figs. 1, 2, 3, 4, and their forms and dimensions are those adopted by "The Engineering Standards Committee." The rolled bars are made in a large variety of sizes, and although the thickness of metal is proportional to the outside dimensions of the sections, it may be varied within small limits for any size of section. The maximum length for rolled bars is about 60 feet.

These bars, in addition to their use for the connecting together of plates, may be used singly or in combination as members of a structure to act as struts or beams.

Fig. 5 shows an angle bar used to connect two plates at right angles to each other. Each plate is secured to one face of the rolled bar by a single line of rivets. It may be here noted that the pitch of the rivets should be a little greater than that used in boiler work. By the use of angle bars and flat plates, girders, of various forms and sizes may be built together. (See the Gusset Stay on Plate XLIV. for an application of the use of *angle bars*.)

Fig. 7 shows the application of a rolled steel bar of beam section in the building up of a compound girder. Two $\frac{3}{8}''$ plates, 8″ wide, are riveted to each flange of the rolled bar, and in this way the flanges, which are to resist tension and compressive stresses when the beam is loaded, are strengthened considerably. The web of the rolled bar is sufficiently strong without the addition of any stiffening plates. The isometric drawing, Fig. 8, shows clearly how the two given views of Fig. 7 are obtained.

For a span of 10 feet this girder will support safely a distributed load of 36 tons—allowing a working tensile strength of 6·4 tons per square inch;—and for any moderate span, say up to 20 feet, the safe distributed load will be inversely proportional to the span.

For example, the safe load for a span of 16 feet $= \frac{10}{16} \times 36 = 22{\cdot}5$ tons.

Here no reduction is made for the slightly increased weight of the girder itself. For the dimensions given on the drawing one foot of length weighs about 76 lb.

EXERCISES.

1.—**Rolled Steel Sections.** Draw *full size* the sections given in Figs. 1 and 2, and $\frac{3}{4}$ *full size* those in Figs. 3 and 4.

2.—**Angle Joint, Fig. 5.** Draw the two given views and add an elevation to the right of the section. *Scale full size.*

3.—**Compound Girder, Fig. 7.** Draw the two given views. *Scale* 8″ = 1 *foot.*

ROLLED STEEL SECTIONS.

TEE SECTION. CHANNEL SECTION. BEAM SECTION.

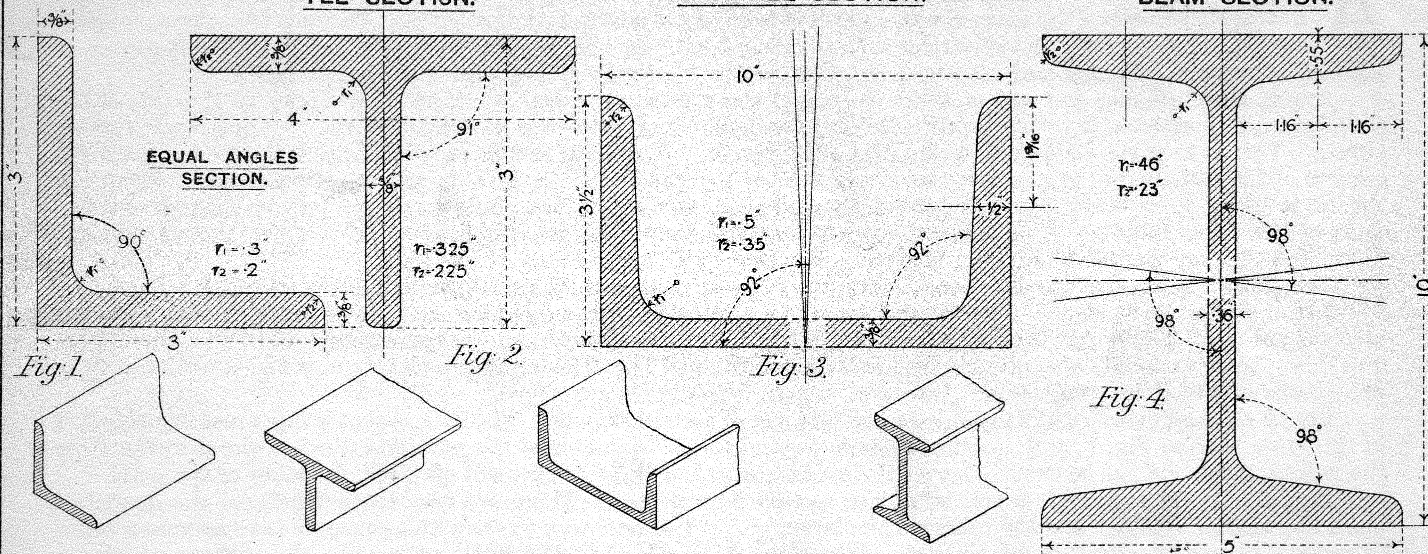


Fig. 1. Fig. 2. Fig. 3. Fig. 4.

ANGLE JOINT.

SECTIONAL ELEVATION

ISOMETRIC VIEWS.

MUST NOT BE DRAWN.

COMPOUND GIRDER.

SECTIONAL ELEVATION

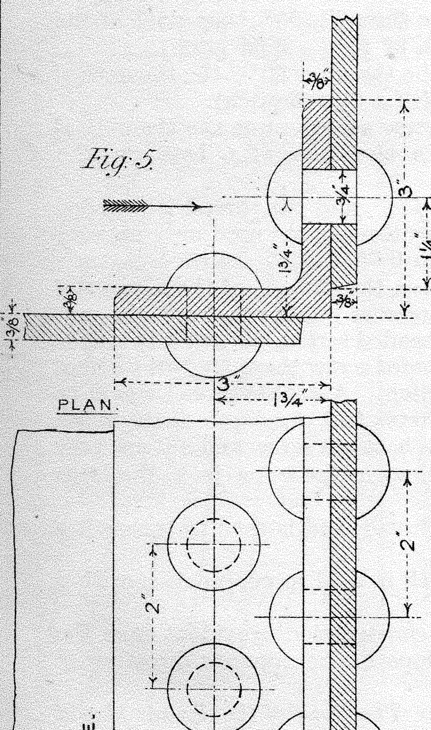


Fig. 5.

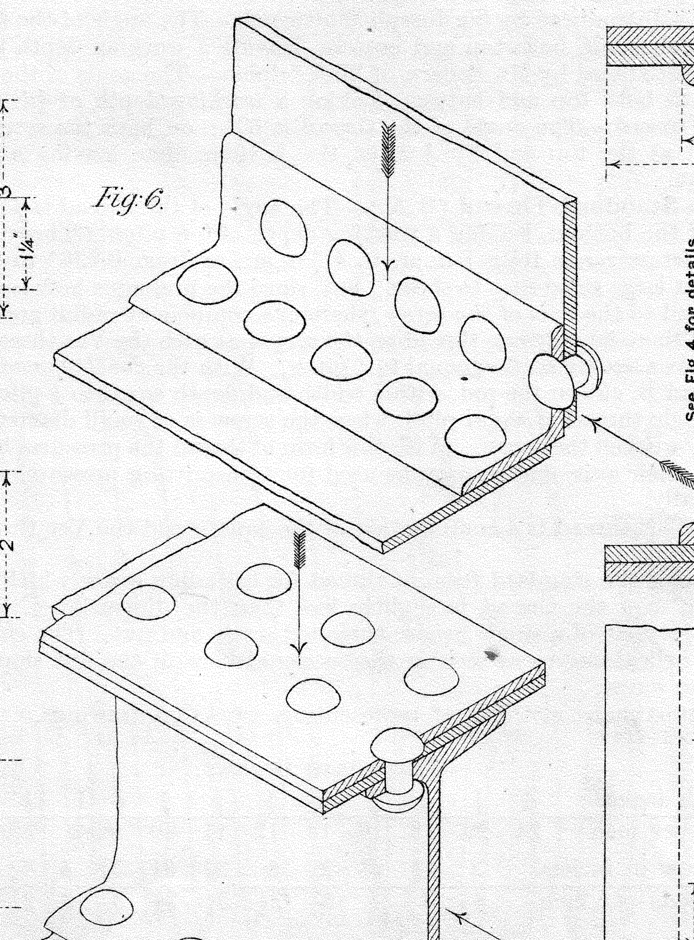


Fig. 6. Fig. 8.

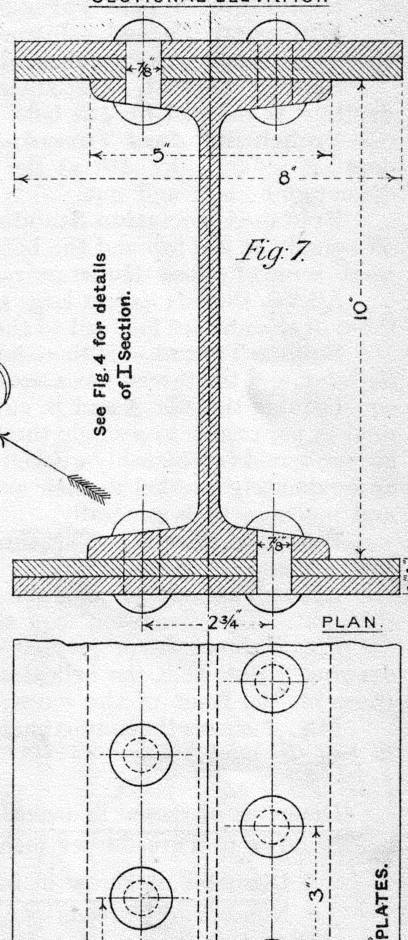


Fig. 7.

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Plate V.—PROJECTIONS AND FORMS OF SCREW THREADS.

A CYLINDRICAL helix is a line of double curvature, that is, it cannot lie on a plane surface, although it becomes a straight line when the surface upon which it is traced is laid flat or developed. To trace this curve, suppose a point to travel upon a cylindrical surface, placed with its axis vertical in such a manner that it moves at a constant angular velocity and also at a constant vertical velocity: the point will trace a **helix.**

Again, if we assume one end of a line to travel along this curve and to be at right angles to the axis of the cylinder in all positions, it will generate a **helical surface,** which is the one generated by the tool in a screw cutting lathe. This is seen the best in a square threaded screw. The slide rest in moving to give the feed, causes the corners of the cutting tool to generate two straight lines at right angles to the axis of the cylinder, upon which the thread is being cut; these lines are carried along, by the threads on the guide screw, uniformly with the revolutions of the lathe spindle. One line generates the helical surface of the right hand side of the thread, and the other line that on the left hand side, the space being cut out by the face of the tool.

The *pitch* of a screw is the distance it will move in the direction of its axis in one revolution through a fixed nut.

Fig. 1 shows the method of drawing the curve. A semicircle is drawn in plan, and the arc divided into a number of equal parts 1-2-3-4, the divisions for the front half, 5-6-7-8, being shown on the same semicircle. The pitch, from 0 to 8, in the elevation, is also divided into eight equal parts. The drawing shows clearly how the elevations of the points are obtained by projection. One and a half revolutions are shown.

Fig. 2 shows a cylindrical wire coiled into the form of a screw thread. The helical centre line must be projected in the same way as Fig. 1, and a series of circles, equal to the diameter of the wire described in the elevation from the points, 1-2-3, &c., as centres. Curves drawn tangential to these circles will give the elevation of the coils.

Fig. 3. In this example a coil of square section is projected. There are two distinct helices, one described upon the smaller cylinder and the other on the larger one. The best way to draw this example is to assume a series of vertical planes passing through the axis of the generating cylinders and cutting the coil; the sections which are all squares have their elevations a series of rectangles. When these elevations are found, the curves which pass through the corners of the rectangles can easily be drawn. The points are shown for one-half revolution.

Sections of Screw Threads.—Fig. 4.

Whitworth Thread.—Introduced by Sir Joseph Whitworth.—The angle of the thread is 55°; one-sixth of the theoretical depth d is rounded off, both top and bottom, leaving a working depth of $\frac{2}{3}d$. (= 0·64 pitch).

Sellers' Thread.—Introduced by Mr. Sellers, of Philadelphia.—The angle of the thread is 60°; the theoretical depth d is reduced by $\frac{1}{8}$th both top and bottom, leaving a working depth of $\frac{3}{4}d$. (= 0·6495 pitch).

French Standard Thread.—The angle of the thread is 60°; on both the screw and the nut the theoretical depth is reduced by $\frac{1}{8}$th at the top and by $\frac{1}{16}$th at the bottom, thus leaving a clearance of $\frac{d}{16}$ between the diameters of bolt and nut.

British Association Standard Thread (B.A.).—The angle of the thread is $47\frac{1}{2}$°, and the thread is rounded off equally at the top and the bottom, leaving a working depth of 0·6 pitch. These threads are used only on very small screws, whose diameters range from 6 m.m. to 1·3 m.m., or from 0·236″ to 0·051″.

All Vee threads have a large resistance to shear; but, since the pressures between the threads of the screw and those of the nut are inclined to the axis of the screw, there is a component radial pressure tending to burst the nut.

Square Thread.—If there be only one thread on the screw, as with the Vee-threaded screw, the screw is *single-threaded,* and the thread has a square section equal to $\frac{1}{2}$ pitch. With the *double-threaded* screw there are two distinct equal square threads A and B, cut on the rod, with a width and depth equal to $\frac{1}{4}$ pitch. A double thread would be used in preference to a single thread of equal pitch when the screw is of small diameter and the depth of the single groove would considerably weaken the screw. With this form of thread the pressures between screw and nut are very approximately parallel to their axis, and so may be used for transmitting pressures as in the screw jack, the lathe and machine tools generally.

Buttress Thread.—This thread is a combination of the square and the Vee thread, and is used to transmit a force in one direction only.

Acme Thread.—This is the standard form of thread for a leading screw which is used in conjunction with a split nut. The dimension a of the thread is slightly less than the dimension b.

Fig. 5 shows the projections of a single square-threaded screw and nut. The construction is evident from the drawing. Note that the helical curves as seen in the sectional view of the nut slope in the opposite direction to those on the front of the screw.

Fig. 7 shows the approximate methods of representing screws on drawings. *The slope of the threads should be half the pitch.*

WHITWORTH SCREWS.

Diameter of Screw in inches	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$
Number of Threads per inch	24	20	18	16	14	12	11	10	9	8	7	7	6	6	5	5	$4\frac{1}{2}$

Diameter of Screw in inches	2	$2\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{3}{4}$	3	$3\frac{1}{4}$	$3\frac{1}{2}$	$3\frac{3}{4}$	4	$4\frac{1}{4}$	$4\frac{1}{2}$	$4\frac{3}{4}$	5	$5\frac{1}{2}$	6
Number of Threads per inch	$4\frac{1}{2}$	4	4	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{4}$	$3\frac{1}{4}$	3	3	$2\frac{7}{8}$	$2\frac{7}{8}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{5}{8}$	$2\frac{1}{2}$

EXERCISES.

1.—**Helix, &c.** Draw Fig. 1: diameter of cylinder, $2\frac{1}{2}$″, pitch 1″. Fig. 2: mean diameter 3″, pitch 2″, and diameter of section $\frac{3}{4}$″. Fig. 3: outside diameter 4″ pitch $1\frac{1}{2}$″, section, a square of $\frac{5}{8}$″ side.

2.—**Sections of Screw Threads.** Draw the various sections of screw threads given in Fig. 4. Pitch in each case to be 1″.

3.—**Square-Threaded Screw and Nut.**—Draw the projections, as in Fig. 4, for an external diameter of screw of 3 inches and pitch $\frac{3}{4}$ inch. *Scale full size.*

4.—**Double-Threaded Left-Hand Screw** (First approximate representation, Fig. 7).—External diameter of screw $2\frac{1}{2}$″, pitch $1\frac{1}{4}$″. *Scale full size.*

PROJECTIONS AND FORMS OF SCREW THREADS.

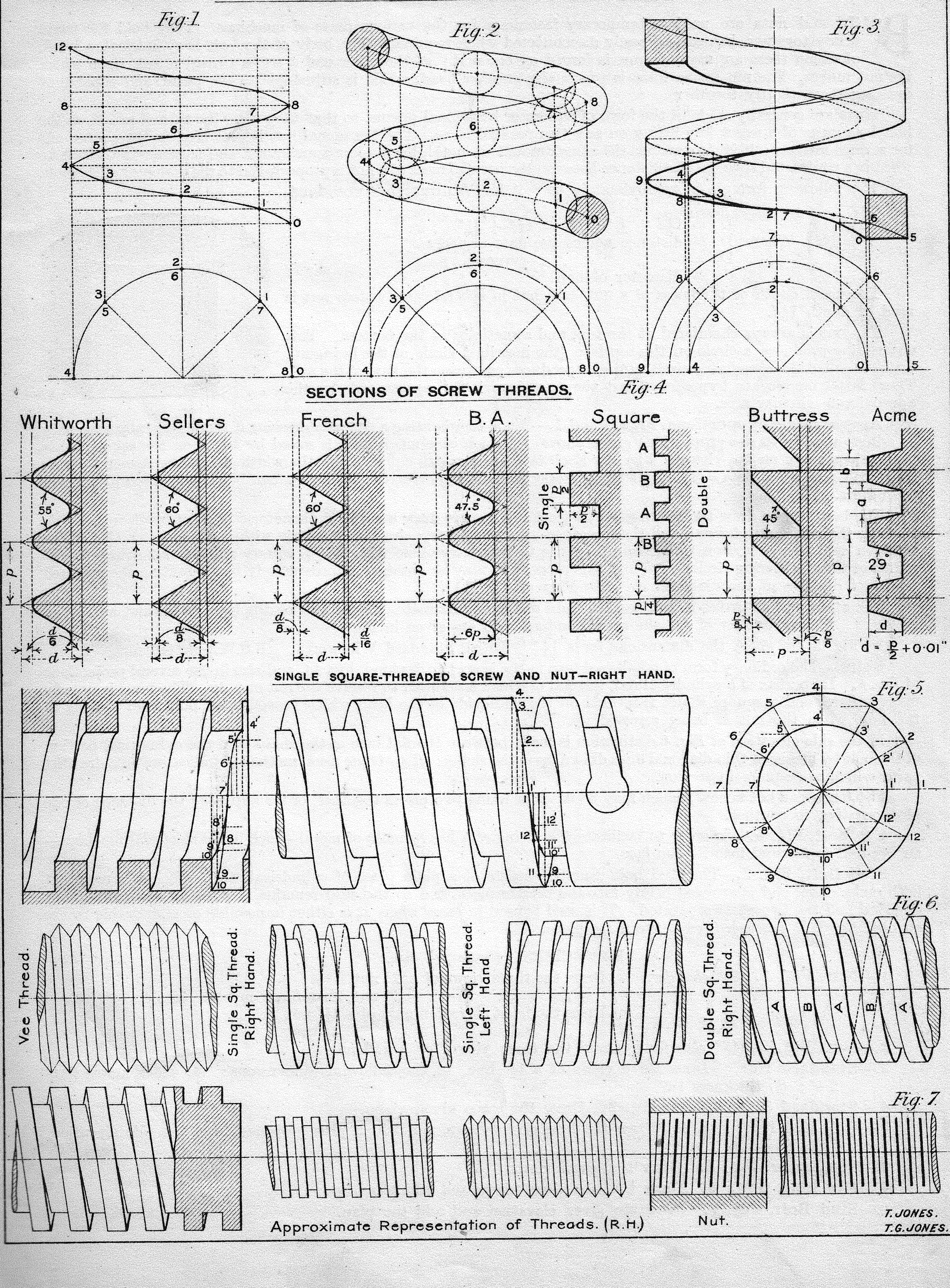
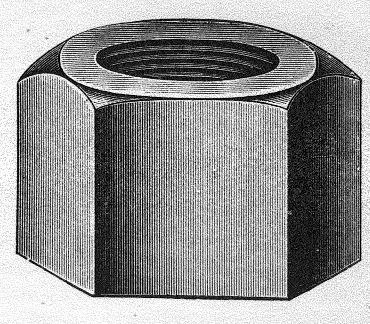


Plate VI.—NUTS AND BOLTS.

BOLTS and nuts are used as temporary fastenings for the various parts of machines. They hold the parts firmly together, but can be easily disconnected when required. The body of the bolt is cylindrical, a head, of which there are many forms, is forged on one end; at the other end a screw thread is cut, upon which the nut moves. The nut forms what is practically another head, which is adjustable to the parts, and capable of drawing them tightly together.

Nuts are generally made of the form of square or hexagonal prisms, so that they may be turned round on the bolt by means of a screw key fitting on any two parallel faces. The hexagonal nut is preferred to the square one for several reasons: it is stronger for the same amount of metal, has a nicer appearance, and is easier to tighten up when in a contracted space. The distance between the parallel faces of a nut is proportional to the diameter of the bolt.



The following formula is approximately correct for **Whitworth's standard nut**:—

$$D = 1\tfrac{1}{2}d + \tfrac{5}{32}''; \quad D_1 = 1\tfrac{3}{4}d + \tfrac{11}{64}''.$$

Where D = distance across the flats or faces.
D_1 = ,, ,, corners.
d = diameter of bolt.

The usual depth or thickness of a nut $= d$, but in special cases a *deep* nut is used of depth $1\frac{1}{2}d$.

The nut is always chamfered at the top, and sometimes at the bottom. This chamfering produces a circle on the top face (the inscribed circle of the hexagon), which is called the *chamfering circle*, and also produces curves on the flats of the nut. These latter curves are hyperbolas, but they are invariably represented by the approximate circular arcs.

Approximate Proportions, Figs. 1 and **2.**—When representing a nut on a drawing it is not generally necessary to adhere strictly to the proportions given above, in which case much time is saved by adopting the approximate proportions given on the views of Fig. 1. Each face has a width equal to d; the width across the corners $= 2\,d$; the width across the flats $= 1\frac{3}{4}\,d$. If these proportions be remembered, each view can be drawn independently of the others.

The best way to draw the hexangonal projection is to first draw a circle of diameter d (the actual clearance hole through the nut is not drawn), and then another—the chamfering circle—concentric with it of diameter $1\frac{3}{4}\,d$. The hexagon can then be drawn by making tangents to the outer circle with the T square and 60° set square. The radius of the chamfering circle is $\frac{7}{8}\,d$, and this proportion may be obtained approximately by estimating the required points of division of the diameter d. (See Plan of Fig. 1.)

The **standard chamfer** is 30°, and the radii of the approximate circular arcs on the faces of the nut and given in Fig. 1. For a chamfer of 45° the corresponding radii are given in Fig. 2

In Fig. 3 are given the dimensions for a $1\frac{1}{4}''$ "British Standard Whitworth" (B.S.W.) slotted nut.

Bolts.—The ordinary form of a bolt is shown in the isometric drawing, Fig. 4a, and also in the several projections of Fig. 4. It consists of a cylindrical rod, the end of which is screwed to receive the nut or nuts. The head is usually made square, and a square shank is formed on the underside of the head which prevents the bolt from turning in the hole when the nut is being screwed up.

In the side elevation of Fig. 4 distinction is made between the flat face of the shank and the cylindrical portion of the bolt by means of thin diagonal lines drawn upon the shank. This is the usual method of indicating on a drawing parts which are square in section.

Fig. 5 shows a **set screw,** which may be used for fixing two pieces together, or for adjusting the distance of one piece relative to another.

Eye Bolt, Fig. 6.—In order to facilitate the lifting of a heavy mass of metal—such as an electrical machine—an eye-bolt is screwed into its top face.

Stud Bolt, Fig. 7.—The stud bolt usually consists of a plain piece of cylindrical steel, which is screwed at both ends. One end is screwed firmly into the thicker piece, and so the stud remains in position when the nut is removed and the two parts separated. This stud is used in cases where it is either impossible or undesirable to use an ordinary bolt *passing through* the main piece.

EXERCISES. (*Scale full size.*)

1.—**Nut, Fig. 1.** (30° *chamfer.*) (*a*) Draw the three views of a $2''$ nut.
(*b*) ,, ,, ,, $\frac{3}{4}''$ nut, chamfered on both sides.
(*c*) Draw the front elevation only of a $1\frac{1}{4}''$ nut.
(*d*) ,, side ,, ,, ,, $1\frac{1}{2}''$ nut.

2.—**Nut, Fig. 2.** (45° *chamfer.*) Draw the three views for a $1\frac{1}{2}''$ nut.

3.—**Standard Nut.** Draw three views of a $1\frac{1}{2}''$ nut, to the following dimensions: (*a*) width across flats $2{\cdot}4''$; (*b*) thickness $1{\cdot}5''$

4.—**Standard Slotted Nut, Fig. 3.** Draw the three given views.

5.—**Bolt and Nut, Fig. 4.** Draw the four given views, and add a plan projected from the side elevation. *Scale full size.*

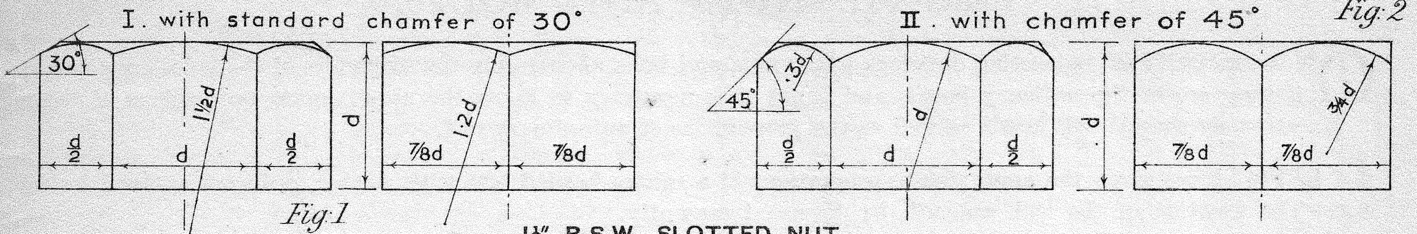
6.—**Set Screw, Fig. 5.** Draw the given views.

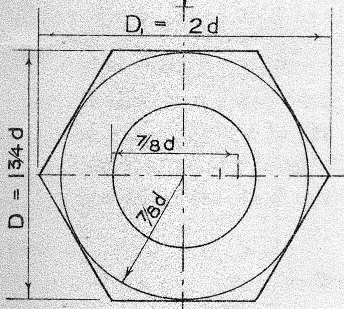
7.—**Eye Bolt, Fig. 6.** Draw the two given views and add the plan.

8.—**Stud Bolt, Fig. 7.** Draw the given elevation and add the plan.

NUTS AND BOLTS.

APPROXIMATE PROPORTIONS FOR ORDINARY NUT.





PROPORTIONS.

d = Diameter of bolt.
D = Width across flats.
D_1 = ,, ,, corners.

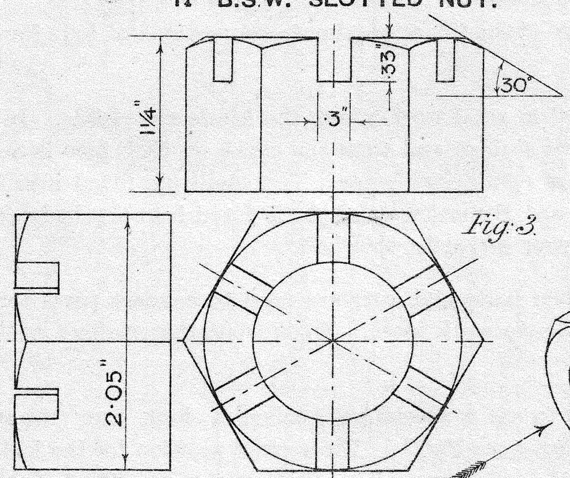
Standard.

$D = \frac{3d}{2} + \frac{5''}{32}$

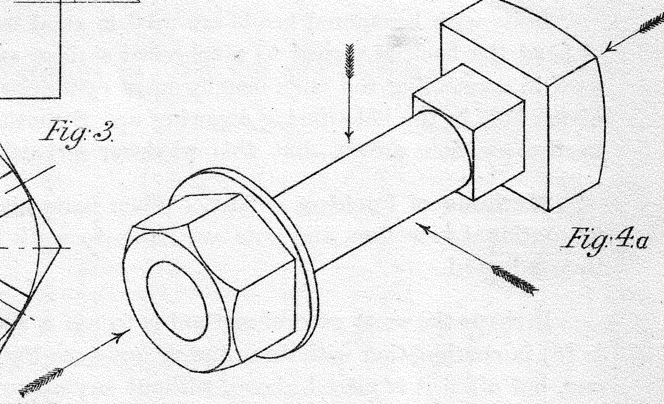
For Drawing purposes—in many cases.

$D = 1\frac{1}{2}d$. $D_1 = 2d$.

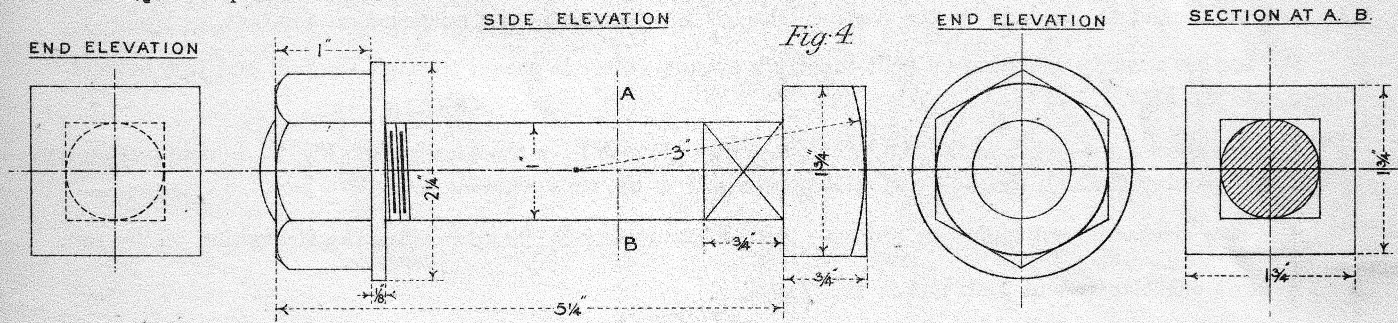
1¼" B.S.W. SLOTTED NUT.



ISOMETRIC VIEW MUST NOT BE DRAWN.

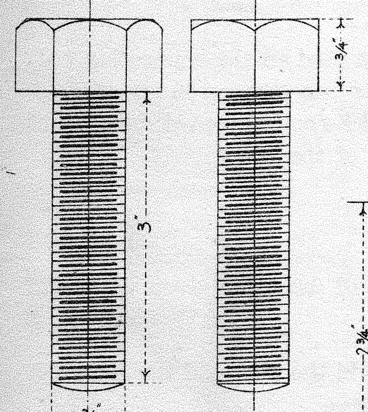


BOLT AND NUT.



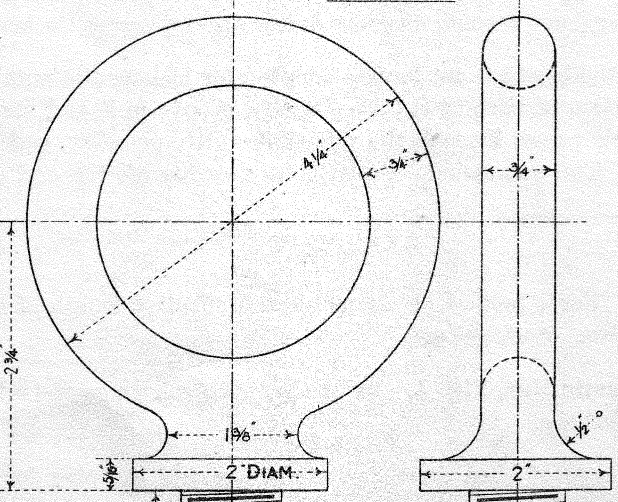
SET SCREW.

ELEVATIONS.



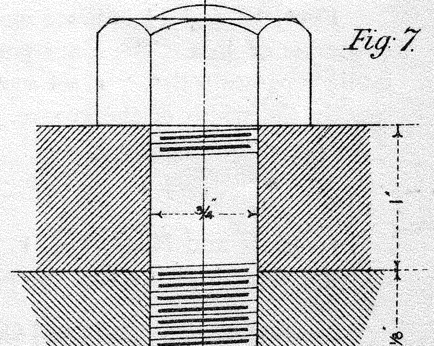
EYE BOLT.

ELEVATIONS.



STUD BOLT.

SECTIONAL ELEVATION



Note.—See Fig. 7, Plate V., for Conventional representation of Screw Threads.

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T. G. JONES.

Plate VII.—NUTS AND BOLTS.

ON the majority of engineering drawings where nuts and bolts appear only the diameters of the bolts are given, if they are of the ordinary forms, and hence it is necessary to know the approximate proportions of the variously shaped bolt heads as well as the proportions of an ordinary nut.

In **Fig. 1** are given the approximate proportions of a square headed bolt with washer, ordinary nut and lock nut. The function of the lock nut will be discussed presently.

Bolt Heads.—In Figs. 3, 4, 5, 6 are given the approximate proportions of four forms of bolt heads which are used for special purposes.

Bolts with hexagonal heads are used in good work where the heads are visible. In some cases with this form of head the back is turned to a spherical surface and then the curve on each face is an arc of a circle. The bolts used in connecting rod ends usually have cylindrical heads. The small pin fitted into the head prevents rotation of the bolt in the hole during screwing up Sometimes the pin is fitted into the body of the bolt—at right angles to the position shown—but this weakens seriously the bolt.

Methods of Locking a Nut.—When bolts and nuts are used to connect parts together which are subjected to continual vibration, the nuts are liable to work loose. To prevent this various methods of locking them have been adopted.

Perhaps the most general method is to use a special nut, called a **lock nut** (the standard thickness of which is $\frac{2}{3}d$) in combination with an ordinary nut—see Fig. 1. The correct position for the lock nut is below the ordinary one, but often it is placed above, without any apparent loss of effectiveness. The lock nut is screwed tightly down and afterwards the ordinary nut, and then by means of two spanners the two nuts are firmly jambed together, and the nuts are locked to the bolt by the friction between the threads in the nuts and on the bolt.

For further security sometimes a split taper pin or split cotter is passed through the bolt end just beyond the upper nut (see Figs. 7 and 8).

A single special nut—such as the B.S.W. **slotted nut** (Plate VI.) or the **Castle nut**, Fig. 2—in conjunction with a split pin passing through the bolt and fitting in a slot in the nut provides a positive lock.

A *spring washer* placed under an ordinary nut assists materially in preventing the slackening of the nut.

A most effective patent lock nut is the *Vislok*.

Fig. 9 shows a method of locking the nuts which are used for fixing the flanges of propeller blades to the propeller boss. The bronze nut is capped and has the washer cast with it. Since the pitches of the small screw and the large one are different, the large nut cannot unscrew unless the set screw slackens back first.

Figs. 10 and 11 indicate methods which are largely adopted for locking the nuts used on the connecting rods of marine engines. The lower portion of the nut is turned with a groove in it and fits into a recess in the plate or into a separate collar. A set screw passes through the side of the plate or collar, and its end is turned down to fit exactly the width of the groove. The nut cannot rise without shearing off the end of the screw.

EXERCISES.

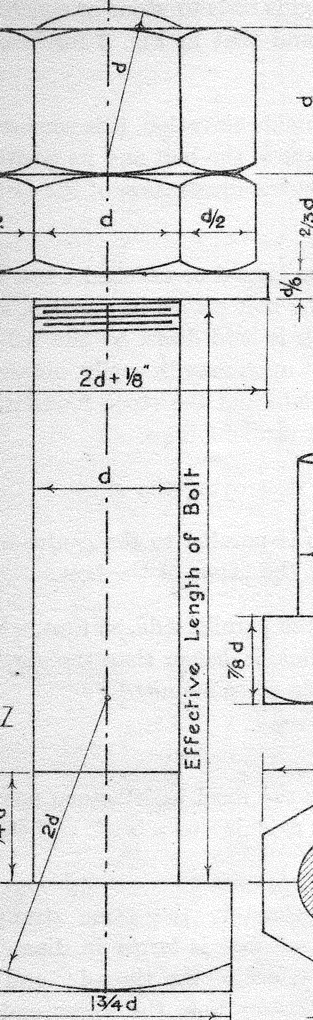
1.—**Bolt and Nuts, Fig. 1.** For a bolt of $1\frac{1}{2}''$ diameter and effective length of $6''$ draw the given view, and add the side elevation and the plan. *Scale full size.*

2—**Automobile Bolt and Castle Nut, Fig. 2.** Draw the two given views and add the plan and the elevation looking on the bolt head. *Scale full size.*

3—**Bolt Heads.** For each type of head draw two elevations and the plan for $d = 1\frac{3}{4}''$. *Scale $\frac{3}{4}$ full size.*

4.—**Various Methods of Locking a Nut.** For each of the examples, Figs. 7, 8, 9, 10, 11, draw three views. For Figs. 7 and 8, *scale full size;* Fig. 9, *scale $\frac{1}{2}$ full size;* Figs. 10 and 11, *scale $\frac{3}{4}$ full size.*

PPROX. PROPORTIONS BOLT, WASHER, IARY NUT & LOCK NUT.



AUTOMOBILE BOLT AND NUT (AMERICAN).

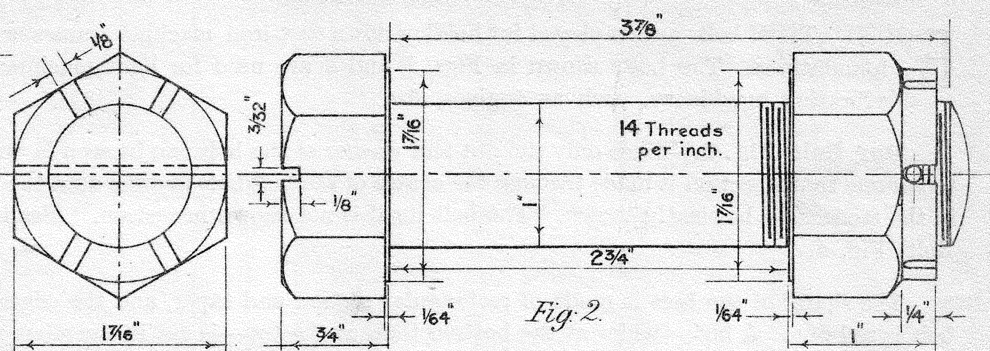


Fig: 2.

APPROXIMATE PROPORTIONS OF VARIOUS BOLT HEADS.

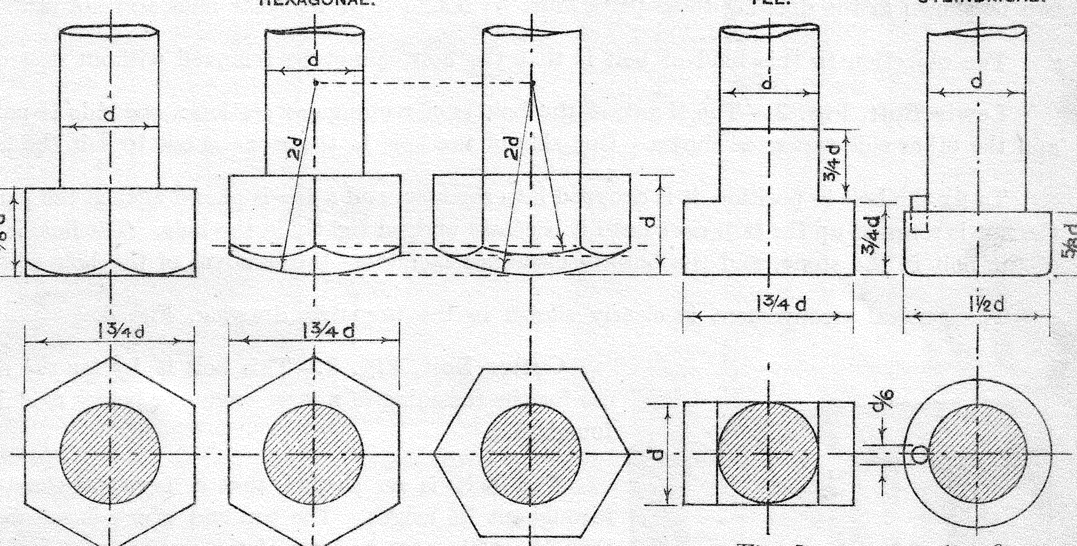
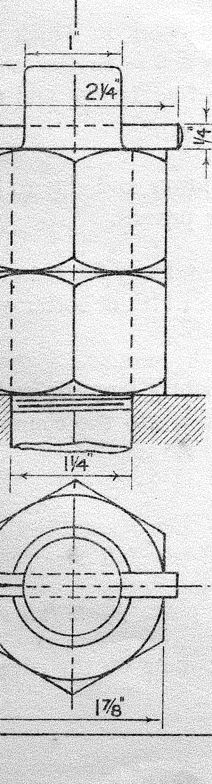


Fig: 3. Fig: 4. Fig: 5. Fig: 6.

OTHER METHODS OF LOCKING A NUT.



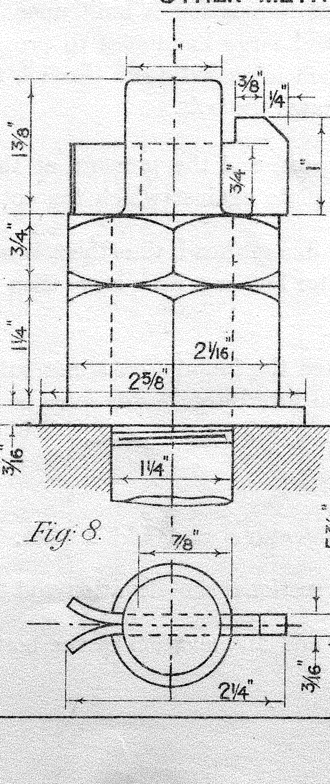


Fig: 8.

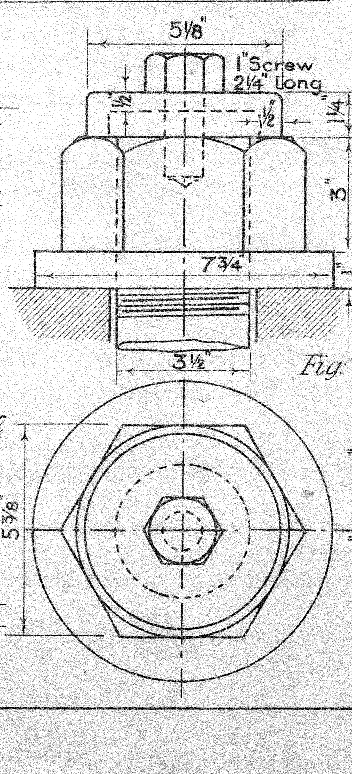


Fig: 9.

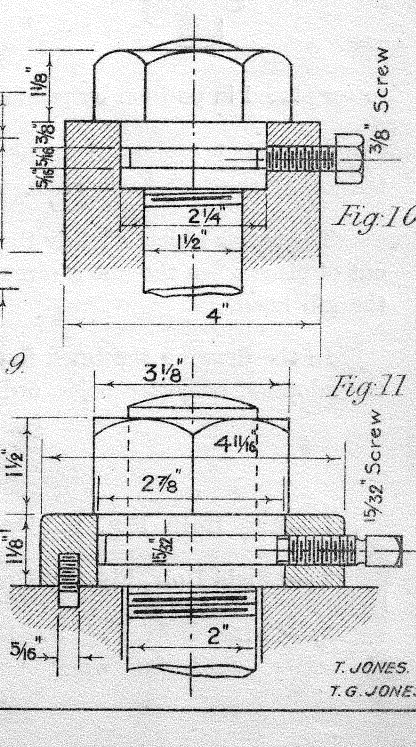


Fig: 10.
Fig: 11

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Plate VIII.—FOUNDATION BOLTS.

FOUNDATION bolts are employed for holding down cast-iron machine frames or engine beds to stone or concrete foundations. The bolts shown in Figs. 1 and 2 are used for light machines, and that in Fig. 3 for heavy vibrating machinery, such as engines, &c.

Rag Bolt, Fig. 1.—Since only the nut and washer of the bolt can be seen in an outside elevation, it is necessary to assume that a section is made through the centre of the bolt hole in order that the shape of the bolt and its position in the stone may be clearly shown. The bolt itself is not shown in section. (See isometric illustration of the Lewis Bolt, Fig. 4.)

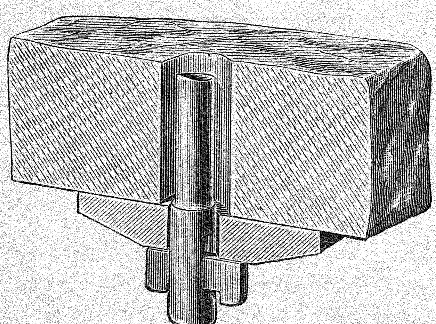
The shank of the bolt is made of rectangular section and taper, and the edges are jagged or notched when the bolt is red hot. A hole—wider at the bottom than at the top—is cut in the stone; the bolt placed in position, and fixed by filling the space round the bolt with molten lead. The cast-iron plate which is held down by the bolt is formed with a circular *facing* round the hole so that a flat surface for the washer to rest upon may be easily obtained by chipping or filing a small area. Since only very small portions of the cast-iron plate and the stone foundation can be shown in the drawing these parts must be bounded by irregular lines and not straight lines.

The objection to this kind of bolt is that the bolt cannot be removed without destroying the stone.

Lewis Bolt, Fig. 2.—The shank of the bolt is of rectangular section: one side is parallel to the centre-line, and the other side tapers as shown. One side of the hole in the stone is cut to suit the taper of the bolt.

To fix the bolt in position, it is dropped into the hole, and a key is placed against the parallel side, so that, when the nut is screwed up the bolt rises until it becomes wedged tightly in the hole. The hole is deeper than the portion of the bolt in the stone, and the bottom clearance facilitates the removal of the bolt when required.

The general arrangement is clearly shown in the isometric drawing, Fig. 4.



Cotter Bolt, Fig. 3.—This bolt is by far the most satisfactory one to use for the fastening of a large engine or heavy machine to a brick and stone foundation.

The bolt is so shaped that it is approximately the same strength throughout its length. The top end which is screwed is larger in diameter than the plain part because of the weakening effect of the thread; and to compensate for the reduction of area by the cotter hole the lower end is enlarged.

The stone or concrete foundation is built upon brickwork in which are left holes for the bolts and large handholes to provide access to the lower ends of the bolts. The bolt is passed down through the foundation, and the cotter placed in position by passing it into the hand hole and through the bolt.

The bolt is tightened up at the top end by means of the nut, and the pressure at the lower end due to the tension of the bolt is distributed by the cast-iron foundation plate against which the cotter presses.

The cotter which is really the bolt head is formed with a double gib head, thus there is no tendency for it to slip out of place when the nut is screwed up. The length of the cotter hole is a little more than the width of cotter over the gib head.

In the drawing the brick foundations are not shown. When the foundation is not sufficiently large to justify the adoption of cotter bolts, ordinary long bolts with plates to form large heads may be built into the concrete.

EXERCISES.

1.—**Rag Bolt, Fig. 1.** Draw the two given sectional elevations and add a plan. *Scale full size*

2.—**Lewis Bolt, Fig. 2.** Draw the given view and add the sectional side elevation and the plan. *Scale full size.*

3.—**Cotter Bolt, Fig. 3.** Draw the two given views and add an elevation of the bolt looking on the edge of the cotter. *Scale full size.*

FOUNDATION BOLTS.

RAG BOLT.

SECTIONAL FRONT ELEVATION   SECTIONAL SIDE ELEVATION

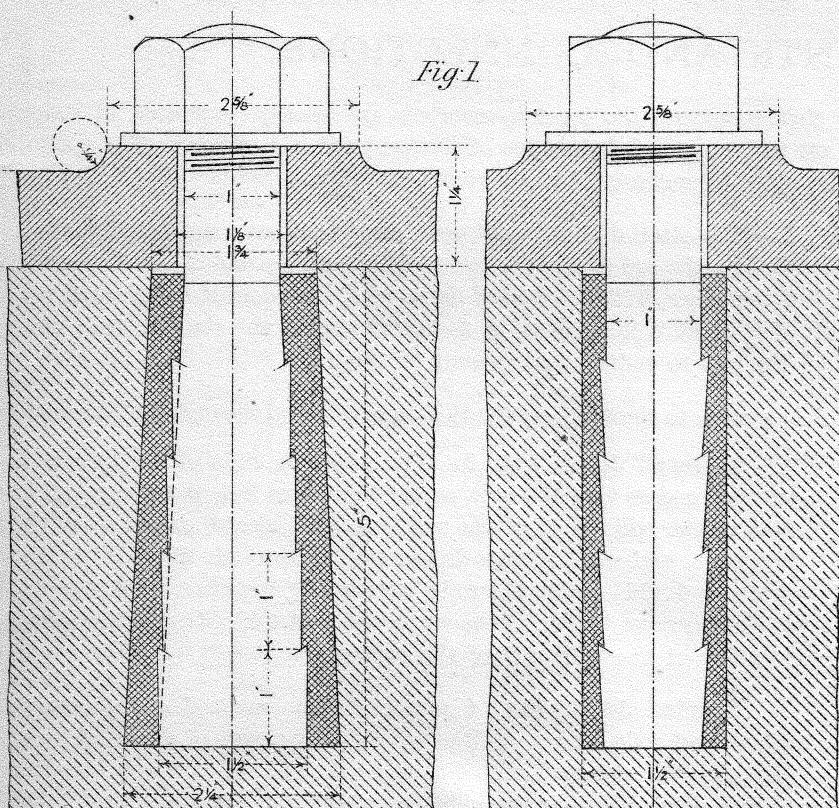
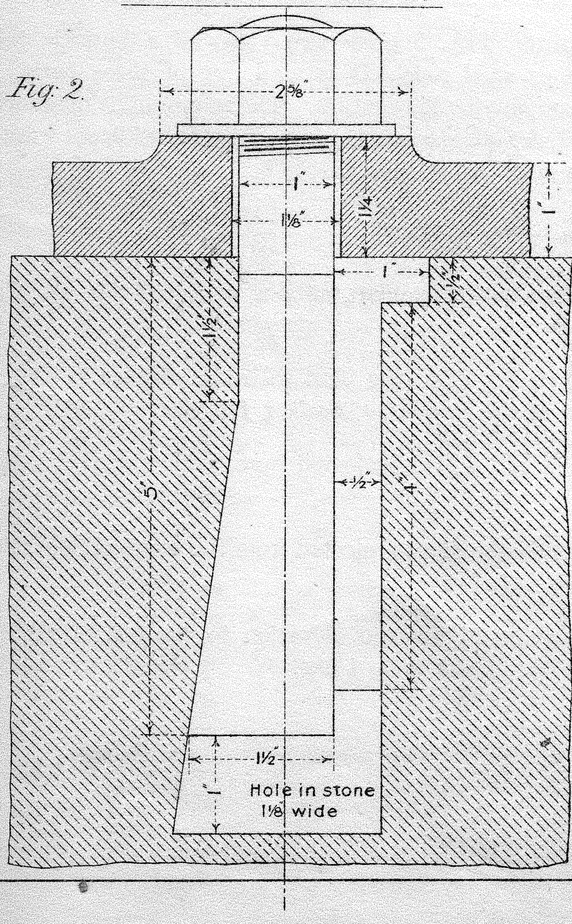


Fig. 1.

LEWIS BOLT.

SECTIONAL FRONT ELEVATION

Fig. 2.



COTTER BOLT.

SECTIONAL ELEVATION

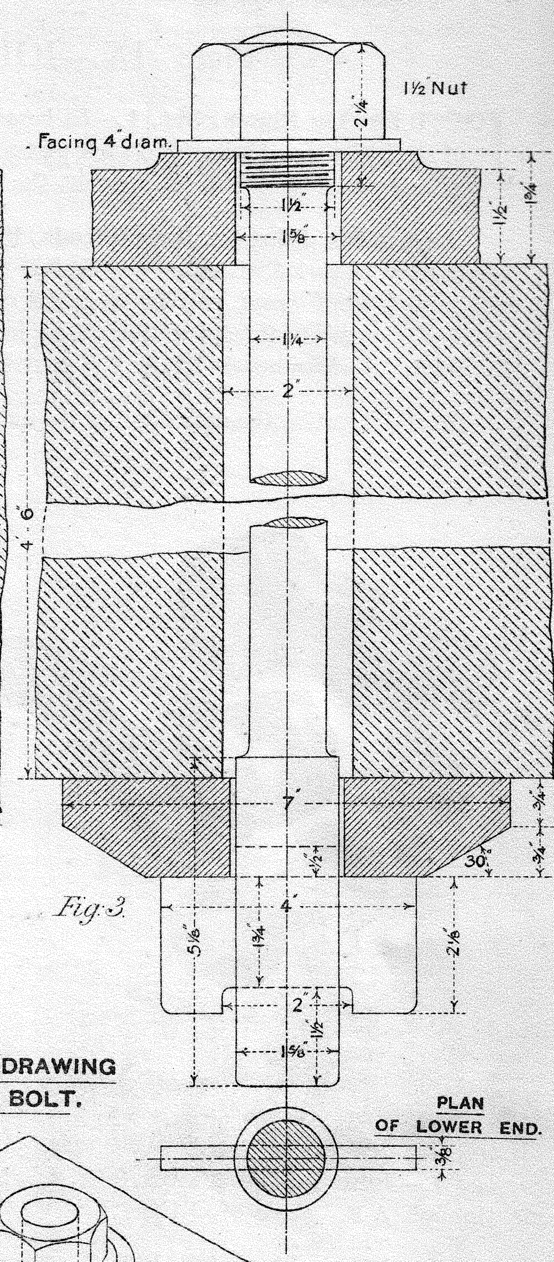


Fig. 3.

PLAN OF LOWER END.

ISOMETRIC DRAWING OF LEWIS BOLT.

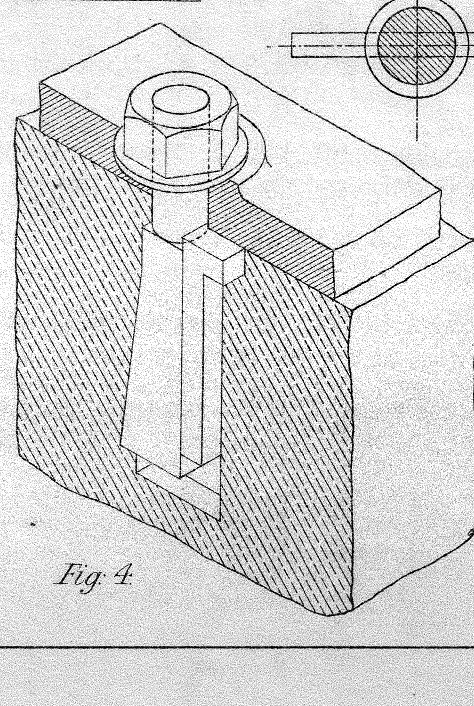


Fig. 4.

This must not be drawn.

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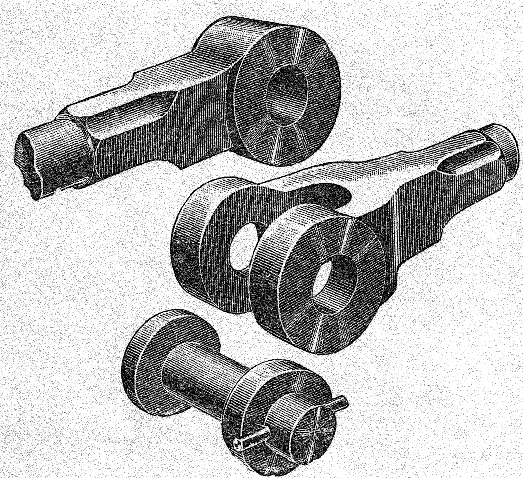
Plate IX.—JOINTS AND CONNECTIONS.

JOINT for Tie Plates, Fig. 1.—In large steel structures the tension members are usually composed of strips of steel plate, and when two such plates are to be connected the type of riveted joint is as shown. The special arrangement of rivets ensures that the tie plate is weakened by one rivet hole only.

Adjustable Joint for Round Rods, Fig. 2.—It is often desirable to have a round rod so constructed that its length may be varied within certain limits. In this case the rod is made in two lengths, and the end of one is screwed with a *right-hand thread,* and the adjacent end of the other, with a *left-hand thread.* A long double-ended steel nut or coupler connects them together. The body of the nut is hexagonal, of the same size as the standard nut used for bolts. A clearance is left in the centre of the nut to reduce the amount of tapping.

In the elevation the upper half of the nut is in section to show clearly the thickness of metal throughout its length.



Cottered Joint, Fig. 3.—The adjacent ends of the two rods are enlarged so that the area across the cotter hole may be equal to that of the solid part of the rod. A steel sleeve passes over the two ends, and steel cotters driven tightly through the slots in the sleeve and rods make the rods butt firmly together. **Clearance,** as shown, must be left in the cotter holes of the rods and the sleeve to allow of the " draw " of the cotters.

Scarfed Joint, Fig. 4 represents one method of connecting round steel rods, such as well pump rods, by means of a scarfed joint. The ends are enlarged and shaped as shown. A cast-iron coupling box or sleeve fits over the joint, keeping the parts in position. The pull or thrust on the rod is transmitted through the shoulders of the joint.

Knuckle Joint. Fig. 5 gives two views of a knuckle joint for wrought-iron or steel rods. It is made up of three parts, as shown in the accompanying illustration. The round pin is made with a head at one end, the other one being formed by a thick loose washer, held in position by a taper pin.

EXERCISES.

1.—**Joint for Tie Plates, Fig. 1.** Draw the given view and add the plan and the sectional side elevation through A B. *Scale* 6″ = 1 *foot.*

2.—**Adjustable Joint, Fig. 2.** Draw the complete outside elevation, the plan with the coupler in section, the given end elevation and the cross section through the centre of the coupler. *Scale* $\frac{3}{4}$ *full size.*

3.—**Cottered Joint, Fig. 3.** Draw the two given views and add a plan projected from the sectional elevation. *Scale* 9″ = 1 *foot.*

4.—**Scarfed Joint, Fig. 4.** Draw the two given views and add the plan, projected from the sectional elevation, and end elevation to the left of the sectional elevation. *Scale* $\frac{3}{4}$ *full size.*

5.—**Knuckle Joint, Fig. 5.** Draw the two given views and add the end elevation to the left of the side elevation, and also the plan projected from the side elevation. *Scale* 9″ = 1 *foot.*

VARIOUS JOINTS AND CONNECTIONS.

JOINT FOR TIE PLATES.

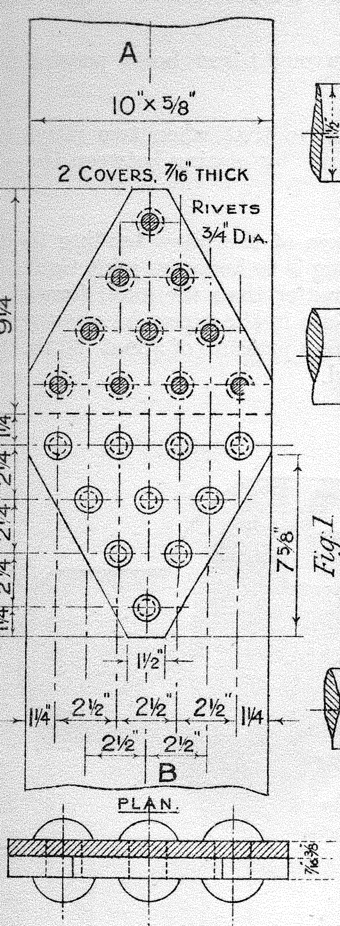


Fig: 1.

ADJUSTABLE JOINT FOR ROUND RODS.

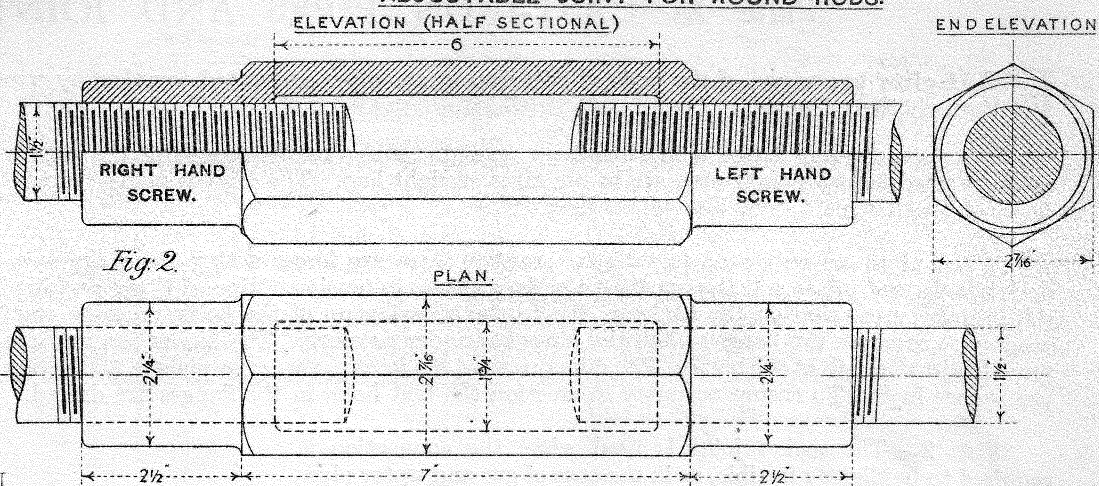


Fig: 2.

COTTERED JOINT FOR ROUND RODS.

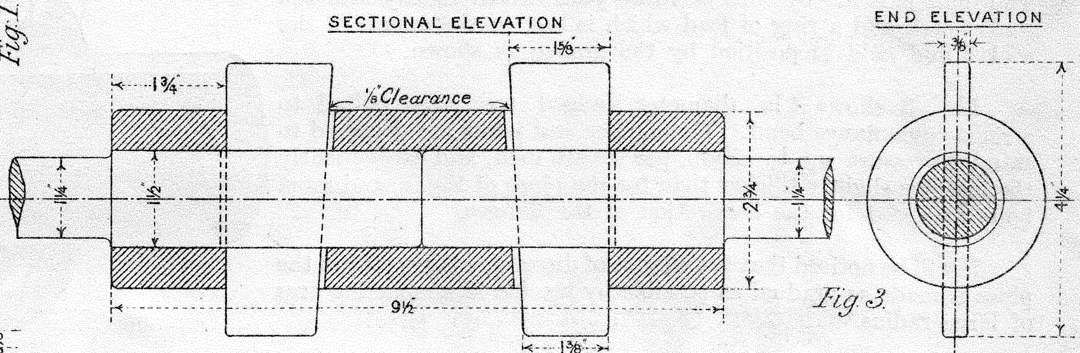


Fig. 3.

SCARFED JOINT.

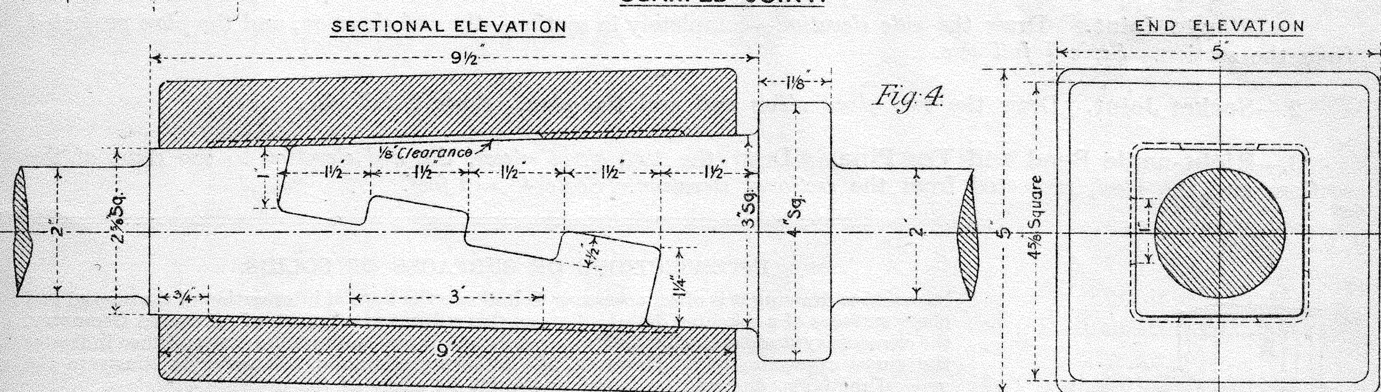


Fig. 4.

KNUCKLE JOINT.

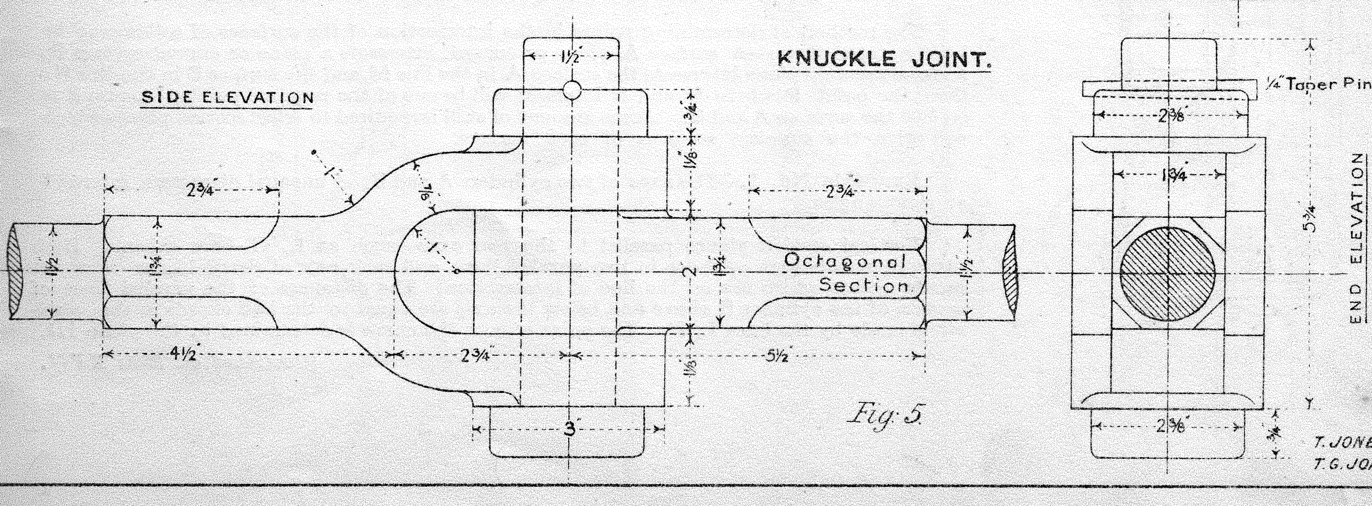


Fig. 5.

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Plate X.—CAST-IRON PIPES AND JOINTS.

FIG. 1 gives two views of the ends of cast-iron steam pipes connected together by wrought-iron bolts passing through their flanges.

The face of each flange is machined up, at right angles to the length of the pipe, so that when two pipes are connected together their axes are in the same straight line. The joint is made " tight " by placing between the faces of the flanges a thin disc of packing.

When pipes are subjected to internal pressure there are forces acting along the axes of the pipes tending to open the flanged joints and thus putting the flange bolts in tension. Hence, if the packing is to keep the joint tight the initial compression on the packing, due to the screwing up of the bolts, must be greater than the axial forces tending to separate the flanges when the pipes are under pressure. The higher the pressure in the pipes the greater must be the strength of the bolts. The dimensions of the joint in this example are those for a pressure of about 60 lb. per square inch. To ensure accuracy in erection the bolt holes in the flanges are drilled.

Fig. 2.—The socket joint is used when the connection is required to be slightly flexible, as in the case of gas and water pipes laid in the ground. A is the spigot end and B the socket end. The joint is made by coils of white yarn driven tightly into the socket end, and a ring of lead which is cast in the mouth of the socket and held in position by the groove as shown.

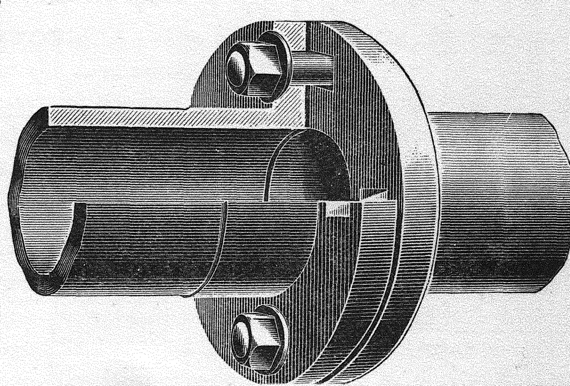


Fig. 3 shows 4 in. diameter flanged tee piece bolted to a right-angle flange bend. These pipes and joints are designed to stand a pressure up to 150 lb. per square inch, and consequently the metal is slightly thicker than for the pipes of Fig. 1, and more bolts are used for the connection of the flanges.

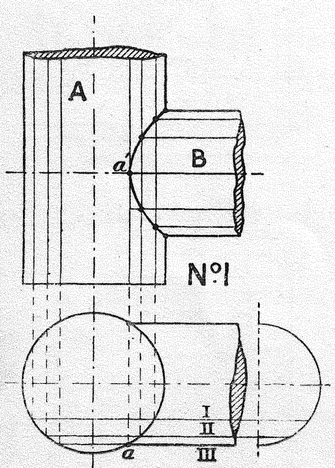
It will be noticed that the change of direction of flow within the pipes is made as gradual as possible by the use of connecting arcs of large radius.

EXERCISES.

1.—**Flange Joint.** Draw the *side elevation*—completely in section—the *end elevation*, and the *plan* projected from the section. *Scale ¾ full size.*

2.—**Socket Joint.** Draw the two *given views* and add the *plan*. *Scale ¾ full size.*

3.—**Right-angle Bend and Tee Pipe.** Draw the two *given views*, the *end elevation* to the right of the section, and the *plan*, projected from the *sectional elevation*. *Scale 4″ = 1 foot.*



INTERSECTIONS OF SURFACES OF SOLIDS.

In machine drawing it is often necessary to indicate the lines of intersection of the curved and plane surfaces of a machine detail; but, as this subject is a branch of Advanced Geometry, the elementary student usually determines only one or two points on a curve and then indicates the curve approximately. However, the following examples will serve to illustrate the general methods, and will be found applicable in many of the drawing exercises.

The method of determining points in the intersection of the surfaces of solids may be briefly stated thus:—A surface A, plane or curved, intersects a plane or curved surface B. Assume a section plane intersects the surface A in the line M, and the surface B in the line N; then, the point P where M and N intersect will be one of the required points, because it is on *both* the surfaces A and B. Some amount of skill is required to select section planes which will give the simplest sections of each surface.

Example No. 1.—The axes of two cylinders A and B, of unequal diameters, intersect at right angles.

Vertical section planes parallel to the two axes—such as I., II.—are taken. Each plane intersects each cylinder in two parallel lines, and each pair of parallel lines intersect in two points which are on the line of intersection. The distances of the parallel lines of section of the cylinder B above and below the axis are equal to the half chords in the semicircle made by the lines I., II. The point a' a on the curve is determined by the plane III.

[*Continued on Plate XVII*,

CAST IRON PIPES AND JOINTS.

SIDE ELEVATION HALF IN SECTION.

FLANGED JOINT.

END ELEVATION

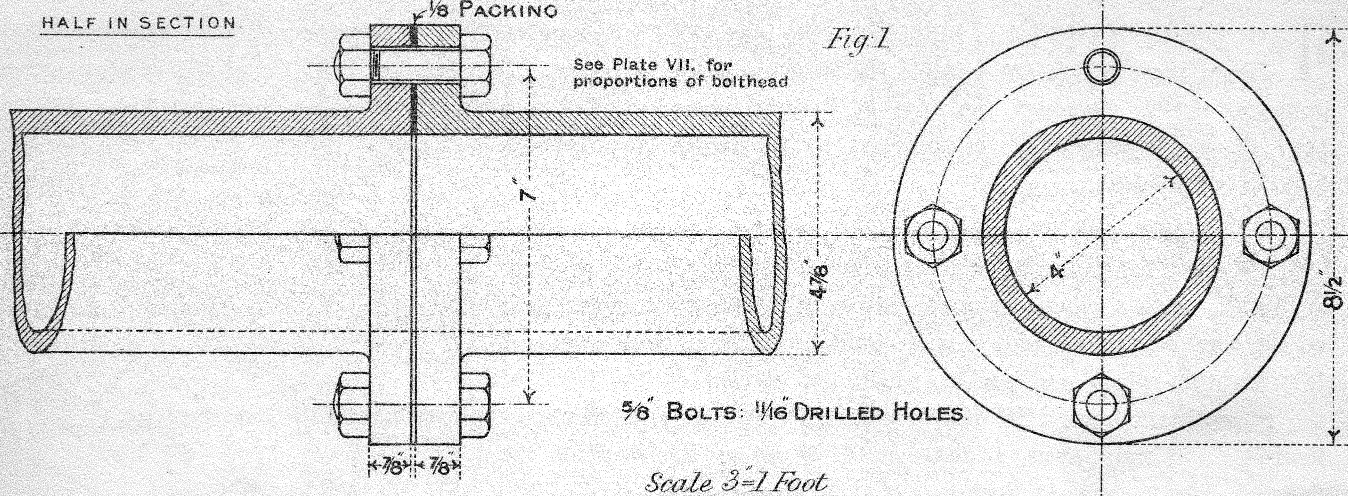


Fig: 1.

Scale 3"=1 Foot

SIDE ELEVATION HALF IN SECTION.

SOCKET JOINT.

END ELEVATION

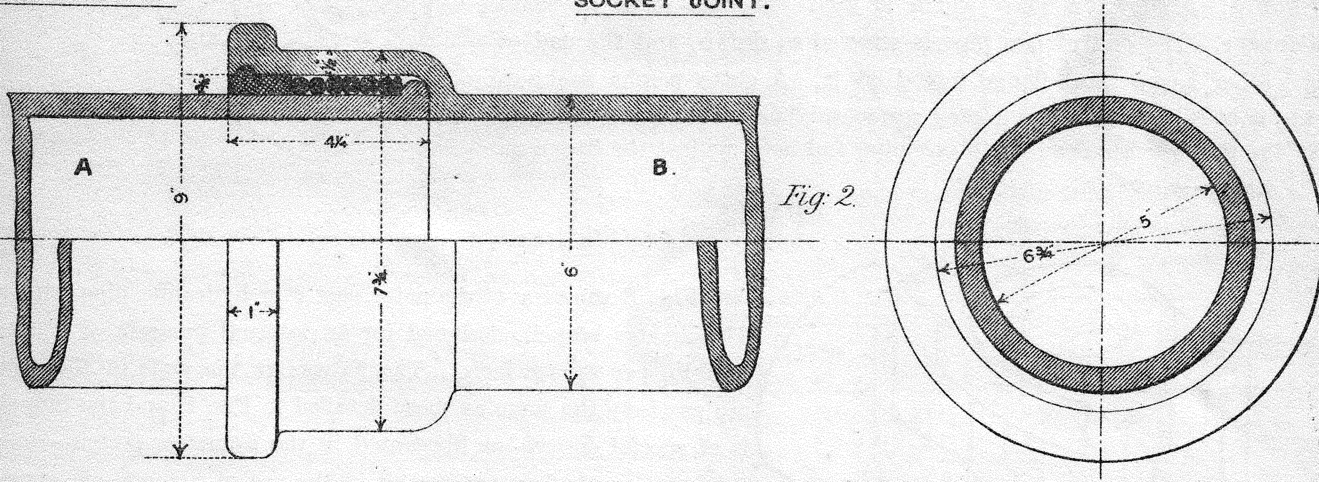


Fig: 2.

4-INCH RIGHT-ANGLE BEND AND TEE PIPE.

END ELEVATION

SECTIONAL SIDE ELEVATION

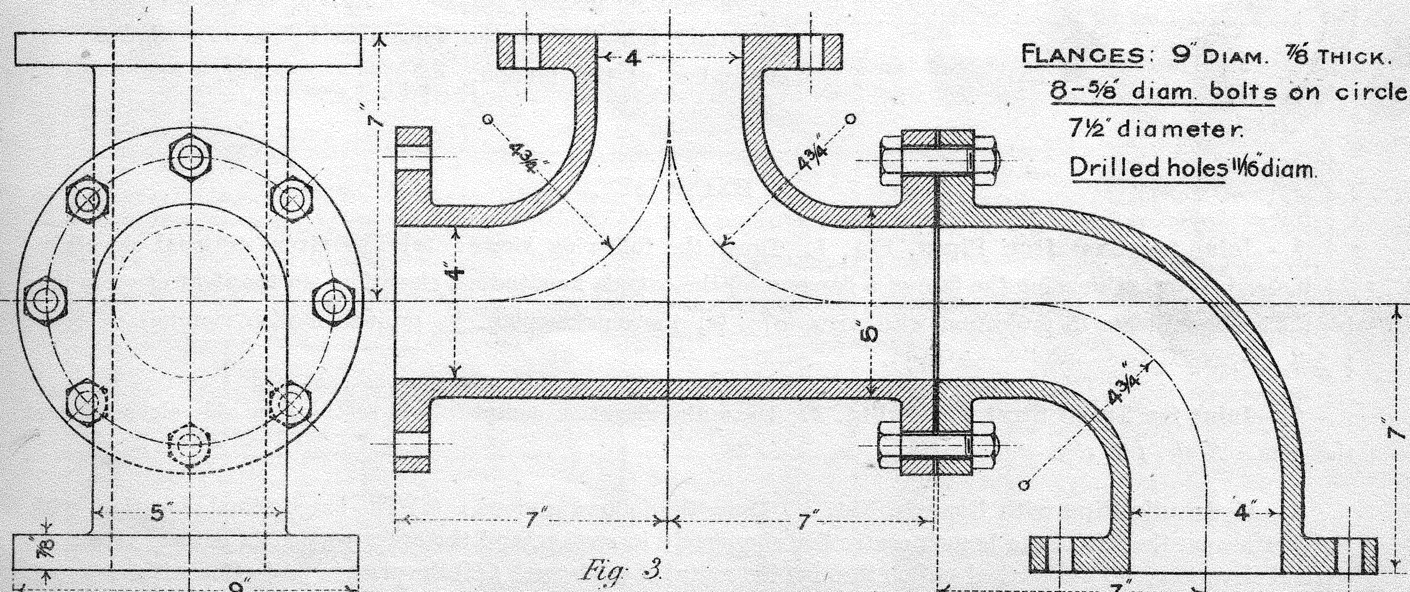


Fig. 3.

Scale 1/5 full size

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Plate XI.—HYDRAULIC PIPE JOINTS.

THE pipes and pipe joints suitable for the purpose of withstanding hydraulic pressure differ widely in design from those which are suitable for ordinary steam pressures. The drawing **Fig. 1** and the accompanying isometric sketch represent the type of hydraulic cast-iron flange joint, for working pressures from 900 to 1200 lb. per square inch, as approved by the *British Engineering Standards Association*.

The flanges are of oval shape and are held together by two $1\frac{1}{2}''$ diameter bolts, which, under the maximum permissible pressure are subjected to a minimum tensile stress of 4·3 tons per square inch on the core area. The joint is made tight by a ring of packing placed between the spigot and socket which are formed on the faces of the respective flanges. It will be noticed that the pipe thickness increases uniformly over a distance of 3″ up to the back of the flange. The increase in thickness of the flange on each side of the bolt hole is to compensate for the weakening effect of the hole.

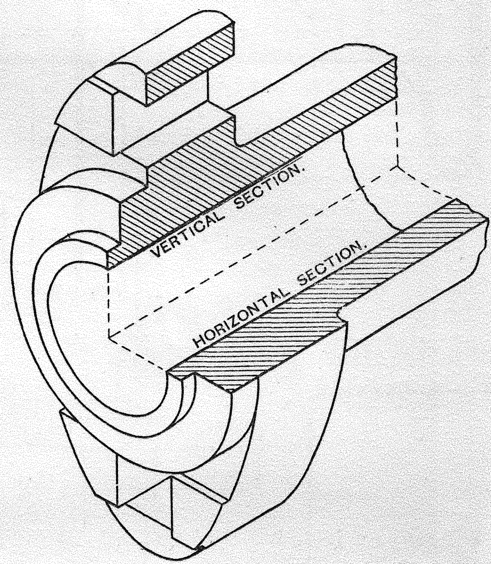


Fig. 2 shows a joint which is used for steel pipes of small diameter. The end of one pipe is screwed as shown, and the end of the other has a small flange forged on it. A gutta percha packing-ring is placed between the jointed surfaces. These are drawn together by the nut which screws on to one pipe and acts against the flange on the other one.

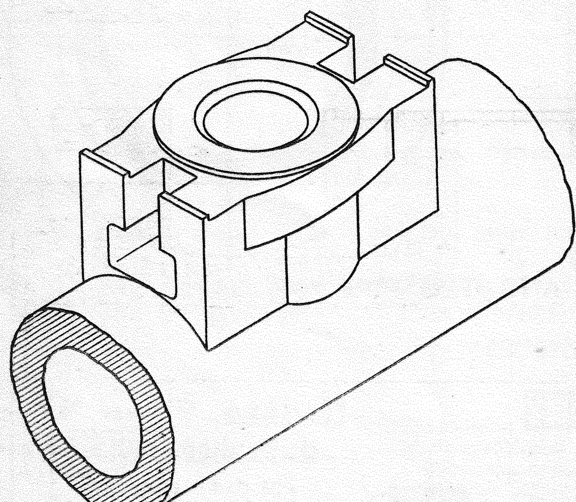


Fig. 3 shows a 4″ diameter cast-iron hydraulic pipe with a 3″ diameter branch, designed for an internal pressure of 900 to 1200 lb. per square inch. The flanges at the ends of the pipe are precisely the same as those detailed in Fig. 1, and the branch is of special design—as illustrated in the isometric sketch—since its short length does not permit of the use of a flange of normal design.

It will be noticed that the thickness of the pipe increases uniformly from $1\frac{1}{4}''$ at the flanges to $1\frac{3}{8}''$ as measured on the centre-line of the branch.

EXERCISES.

1.—**Joint for Cast Iron Pipes, Fig. 1.** Draw the following views: (*a*) The given sectional elevation; (*b*) the end elevation showing the face of a flange; (*c*) the outside longitudinal elevation—to the right of view (*b*); (*d*) the sectional plan, in projection with view (*a*); (*e*) the outside plan, in projection with view (*c*). *Scale, $\frac{1}{2}$ full size.*

2.—**Joint for Small Steel Pipes, Fig. 2.** Draw the elevation completely in section, the side elevation and the plan. *Scale, full size.*

3.—**Hydraulic Pipe with Branch, Fig. 3.** Draw the following views: (*a*) The longitudinal elevation, with the portion on the left of the branch centre line completely in section, and the other portion in outside view; (*b*) the end elevation; (*c*) the vertical section through the centre of the branch; (*d*) the plan, in projection with view (*a*). *Scale, $\frac{1}{2}$ full size.*

HYDRAULIC PIPE JOINTS.

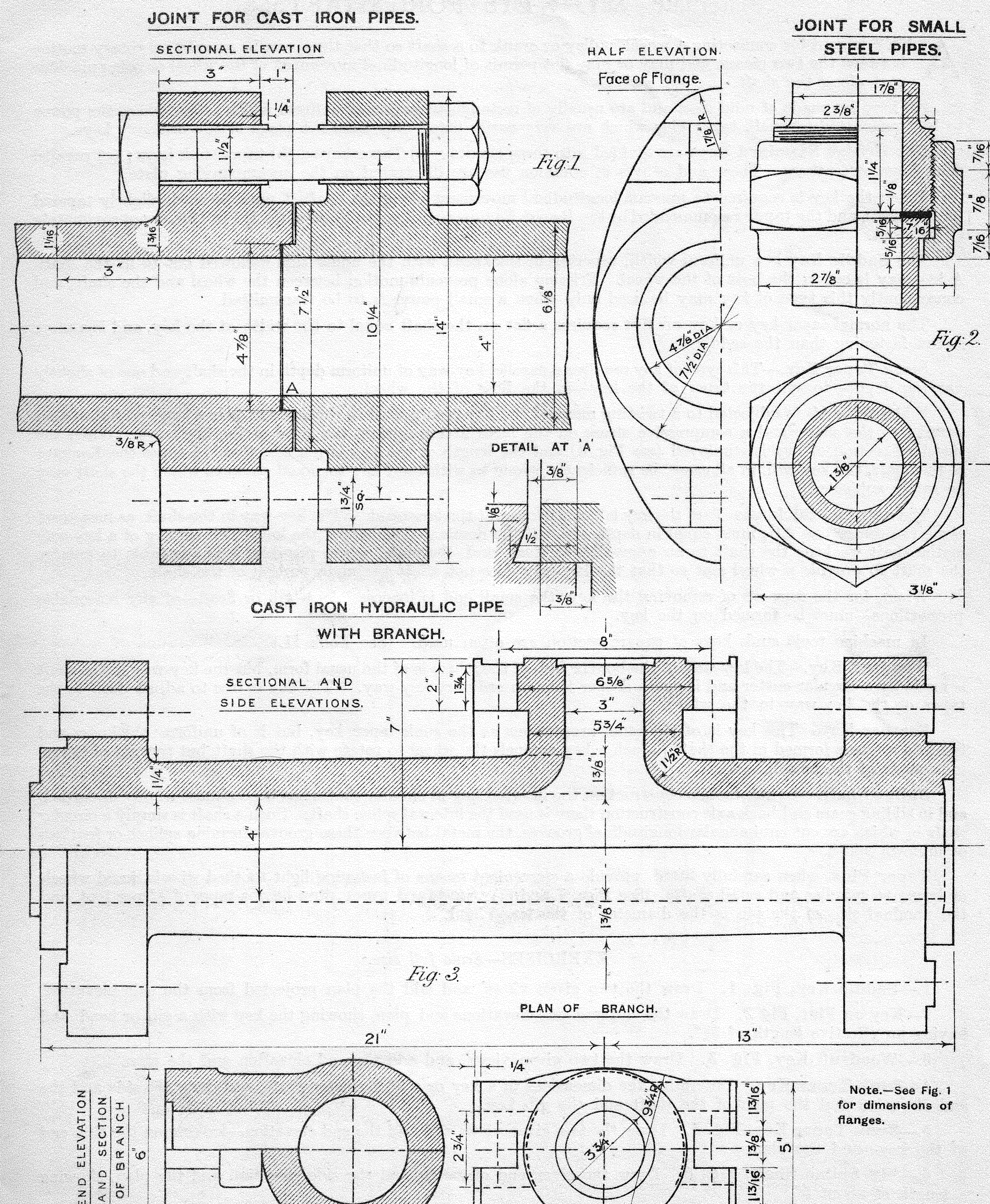


Plate XII.—KEYS FOR SHAFTS.

A KEY is used for connecting a wheel, pulley or crank to a shaft so that there can be no relative rotary motion between the two pieces, and may or may not permit of longitudinal movement of the wheel or other machine part along the shaft.

All keys are made of mild steel and are usually of rectangular section of uniform width ; but, where the power to be transmitted is small, taper pins which are very easily fitted may take the place of the ordinary keys.

The **British Standard Keys** are divided into three classes : (a) Taper keys ; (b) taper sunk keys ; (c) parallel sunk keys or feathers. These and others in common use are illustrated on the accompanying plate.

When the key is required to prevent longitudinal movement along the shaft it must be made slightly tapered in thickness, and the **taper** recommended by the *British Engineering Standards Committee* is **1 in 100**, or approximately $\frac{1}{8}''$ per foot.

The **Saddle Key** is a uniform width, tapering in thickness, with the under side hollowed out to fit the shaft. A key-way is cut in the boss of the wheel. Friction alone prevents motion between the wheel and the shaft, and consequently this type of key may be used only when a small power is to be transmitted.

The normal taper key or **key on flat** requires a flat on the shaft equal to the width of the key, and is a more secure fastening than the saddle key

Sunk Taper Key.—This type of key requires a parallel key-way of uniform depth in the shaft and one of slightly varying depth—to suit the taper of the key—in the boss of the wheel.

When the shaft is subjected to a twisting moment the key is subjected to shear stress over a longitudinal section parallel to the shaft, and a compressive stress on the faces fitting against the sides of the key-ways. When the standard section of key is employed (see Fig. 4) and the length of the key is not less than $1\frac{1}{2}$ times the diameter of the shaft, the key will, in all cases, be sufficiently strong to withstand any torsional stress to which the shaft may be safely subjected.

The **nominal thickness** T of the key is the thickness at the large end. The key-way in the shaft, as measured along the centre line AB, must equal in depth one-half the nominal thickness of the key. The cutting of a key-way in the shaft weakens the shaft to an appreciable extent, and, therefore, where possible, it is advisable to enlarge the shaft to provide a wheel seat so that the key-way does not affect the main portion of the shaft.

When, for the purpose of removing the key, the small end is inaccessible, a gib or head—of any reasonable proportions—must be formed on the key.

In machine tools sunk keys of square section are often used. (See Book II.)

Woodruff Key.—The key-way in the boss or hub of the wheel is of the usual form, but the key-way in the shaft is made by a circular cutter and is much deeper than an ordinary key-way. The key is free to adjust itself to the taper of the key-way in the wheel.

Feather Key.—This key is of the same proportions as the sunk taper key, but is of uniform thickness and fits into a recess formed in the shaft. Such a key compels the wheel to rotate with the shaft, but permits of lateral movement of the wheel.

Spline Shaft.—In automobile construction the inserted key in circular shafts has been almost wholly discarded, and in sliding gears and back axle construction there is used the integral spline shaft. Such a shaft is simply a circular shaft in which are cut equi-angular longitudinal grooves, the metal between these grooves forming splines or feathers of uniform width.

Taper Pins, when carefully fitted, provide a convenient means of fastening light toothed wheels, hand wheels or levers to spindles and small shafts. (See Figs. 7 and 8.) **Standard taper pins** have a taper of $\frac{1}{4}''$ per foot, and the nominal size of the pin is the diameter of the larger end.

EXERCISES—*Scale full size.*

1.—**Saddle Key, Fig. 1.** Draw the two given views, and add the plan projected from the side elevation.

2—**Key on Flat, Fig 2.** Draw the end and side elevations and plan, showing the key with a gib or head and having an effective length of $3\frac{3}{4}''$.

3.—**Woodruff Key, Fig. 3.** Draw the two given views, and add the end elevation and the plan.

4—**Sunk Taper Key.** Determine the dimensions of a key or a 3″ diameter shaft, and draw the side and the end elevation and the plan of the shaft and the gib key.

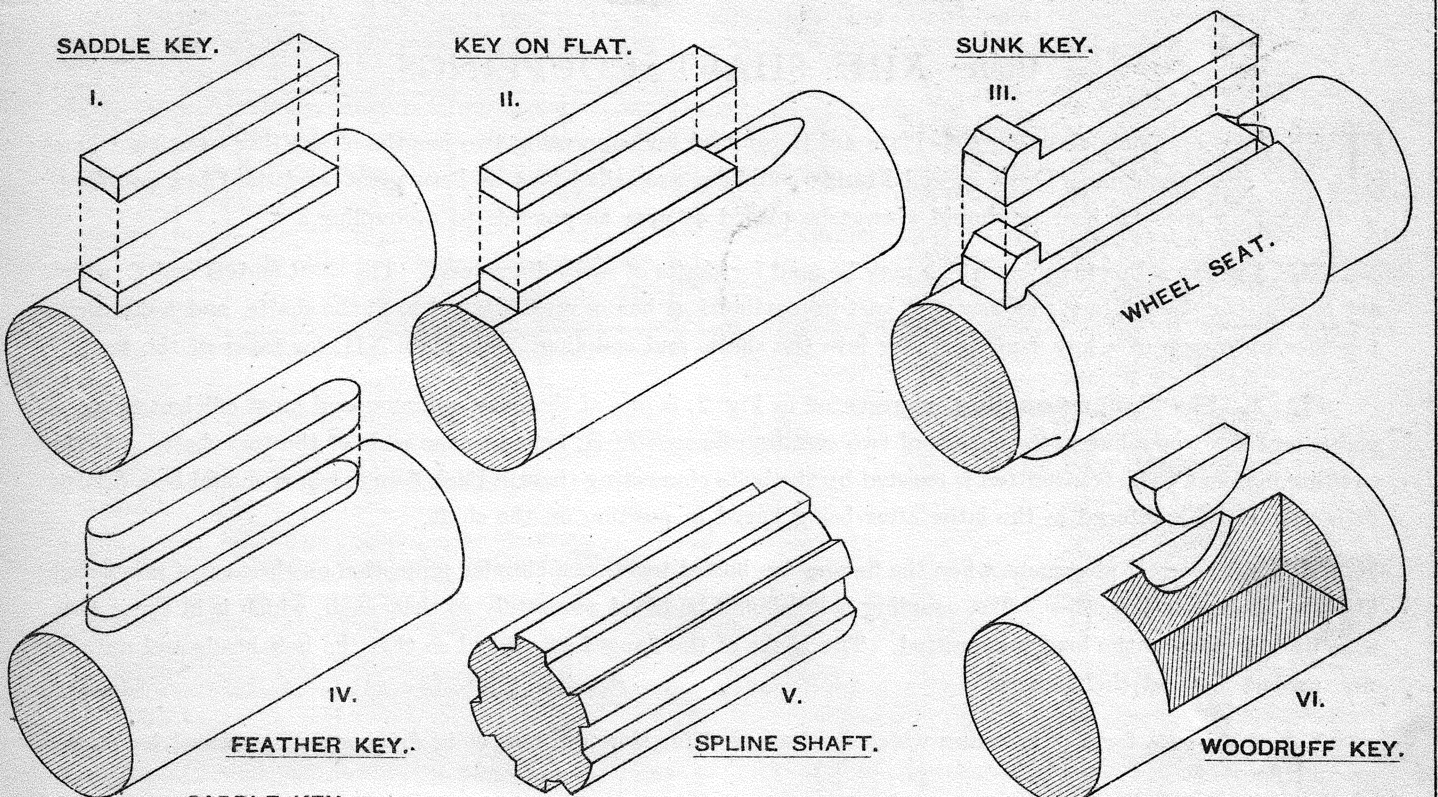
5.—**Sunk Taper Key, Fig. 5.** Draw the two given views and add the end elevation—looking on the thin end of the key—and the plan.

6.—**Six Spline Shaft, Fig. 6.** Draw the given end elevation and the side elevation and the plan, showing a length of about 5″.

7.—**Pin Keys, Figs. 7 and 8.** Draw the given views.

KEYS FOR SHAFTS.

ISOMETRIC VIEWS—FOR REFERENCE ONLY.



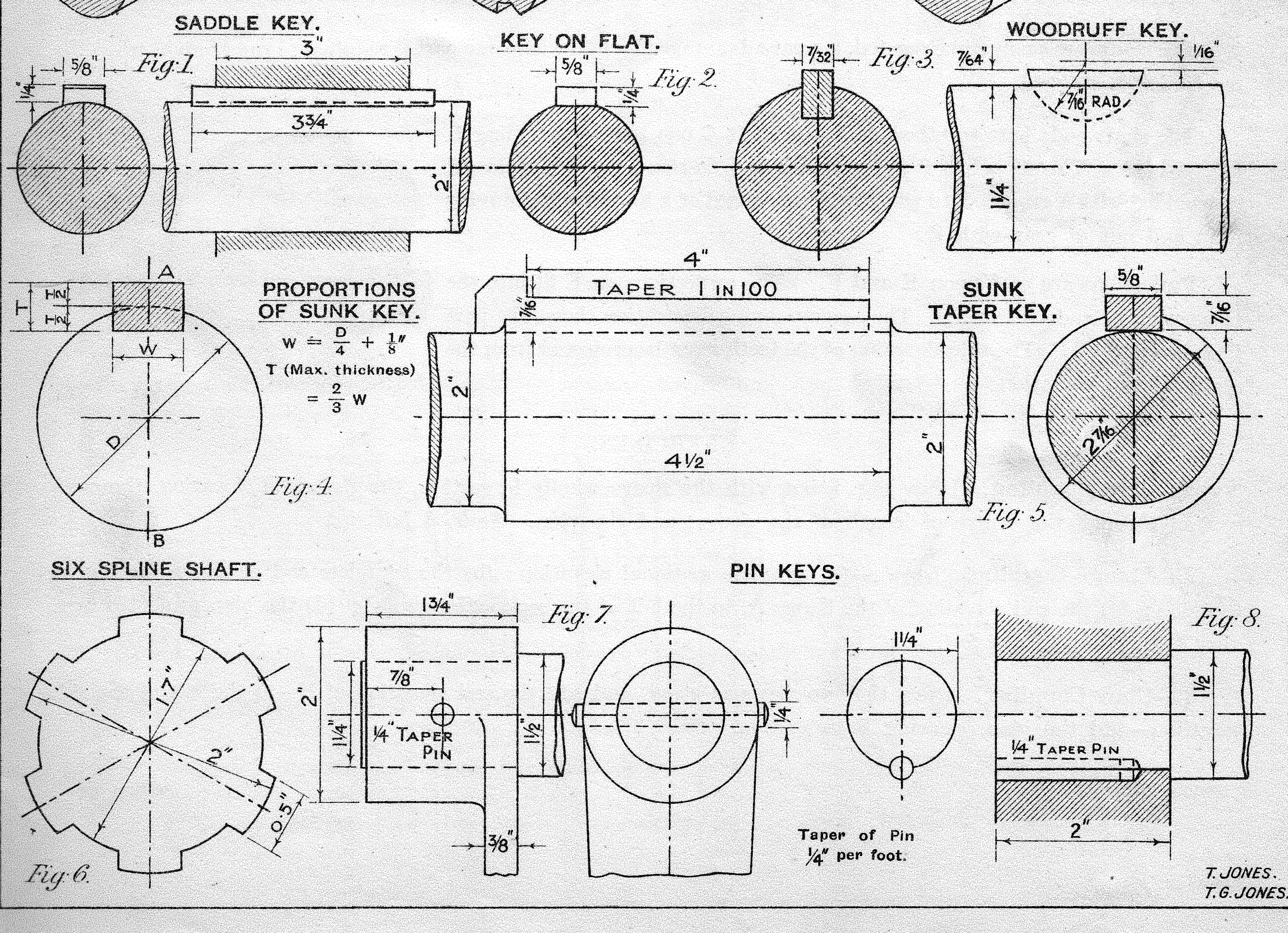


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Plate XIII.—SHAFT COUPLINGS.

THE three couplings given on this Plate are those used for connecting two lengths of shafting together whose axes are coincident. The **Box** and **Flange** couplings are called Fast or Permanent, and the **Claw** coupling, a disengaging one. A bearing should always be placed as near as possible to a coupling.

Fig. 1 shows a Box coupling which may be used for shafts of small diameters. The shaft ends B and C, which are enlarged or *bossed*, butt together. A cast-iron cylindrical box or muff is bored to fit the shafts, and holds them together by means of a key sunk half way into the shafts and coupling. See Plate XII. for taper of the key.

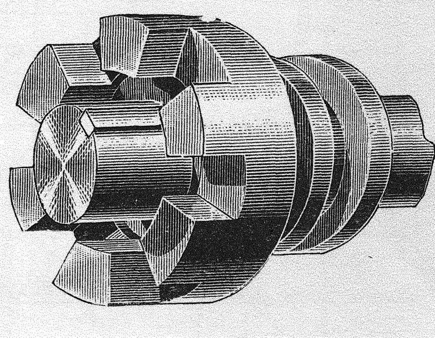
Fig. 2.—The **Flange Coupling,** represented in Fig. 2, is one of the most common and most efficient types of permanent shaft couplings. It consists of two cast-iron flanges A, B, keyed to the ends of the two shafts, and the twisting moment to be transmitted is resisted by the bolts connecting them. Each flange, which should be a driving fit on the shaft is refaced in the lathe after being keyed in position on the shaft.

To ensure correct alignment when the flanges are bolted together a circular projection on the face of one flange fits into a recess in the other. The coupling bolts should be fitted accurately so that each, which is in shear, may take its full share of the load transmitted. The backs of the flanges are dished so that the bolt heads and nuts do not project beyond their edges.

Fig. 3 shows a form of coupling which is often used for slow running main driving shafts placed below the ground floor.

The shafts are easily disconnected, a great advantage in a mill where one part is required to run at night when the remainder is stopped.

The shaft ends butt together. On the shaft C one half the coupling E is keyed fast. The other half F slides upon D on a feather key. The piece F is provided with a groove, into which the forked end of a lever fits for moving it in and out of gear with E.



Projections are cast upon E and F. The projections on E fit into the recesses on F, and vice versa. The accompanying illustration shows the part F on the shaft D. The side elevation of the teeth must be projected from the sectional end elevation.

EXERCISES.

1.—**Box Coupling.** Draw Fig. 1 but with the sleeve wholly in section, the given end elevation, the cross section through the centre of length of the sleeve, and the plan. *Scale, $\frac{3}{4}$ full size.*

2.—**Flange Coupling.** Draw :—(a) complete sectional elevation ; (b) the complete end elevation looking on the nuts ; (c) the end elevation of the flange A, to the left of the sectional view (a) ; (d) the plan projected from view (a). *Scale,* 6″ = 1 *foot.*

3.—**Claw Coupling.** Draw the two views as given, and add the end elevation of E—to the left of the side elevation—and the plan, showing F in section. *Scale,* 3″=1 *foot.*

SHAFT COUPLINGS.

BOX COUPLING.

SIDE & SECTIONAL ELEVATIONS

END ELEVATION

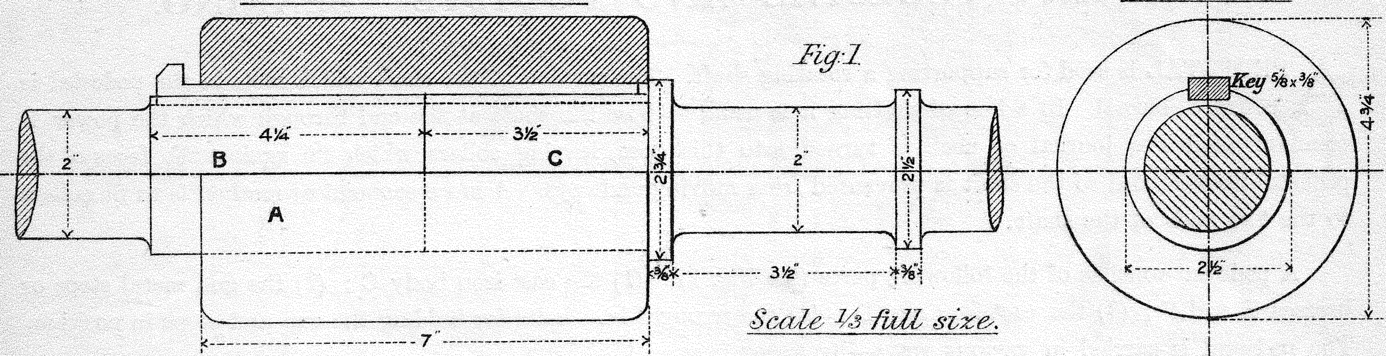


Fig: 1.

Scale ⅓ full size.

FLANGE COUPLING.

SIDE & SECTIONAL ELEVATIONS

END ELEVATIONS

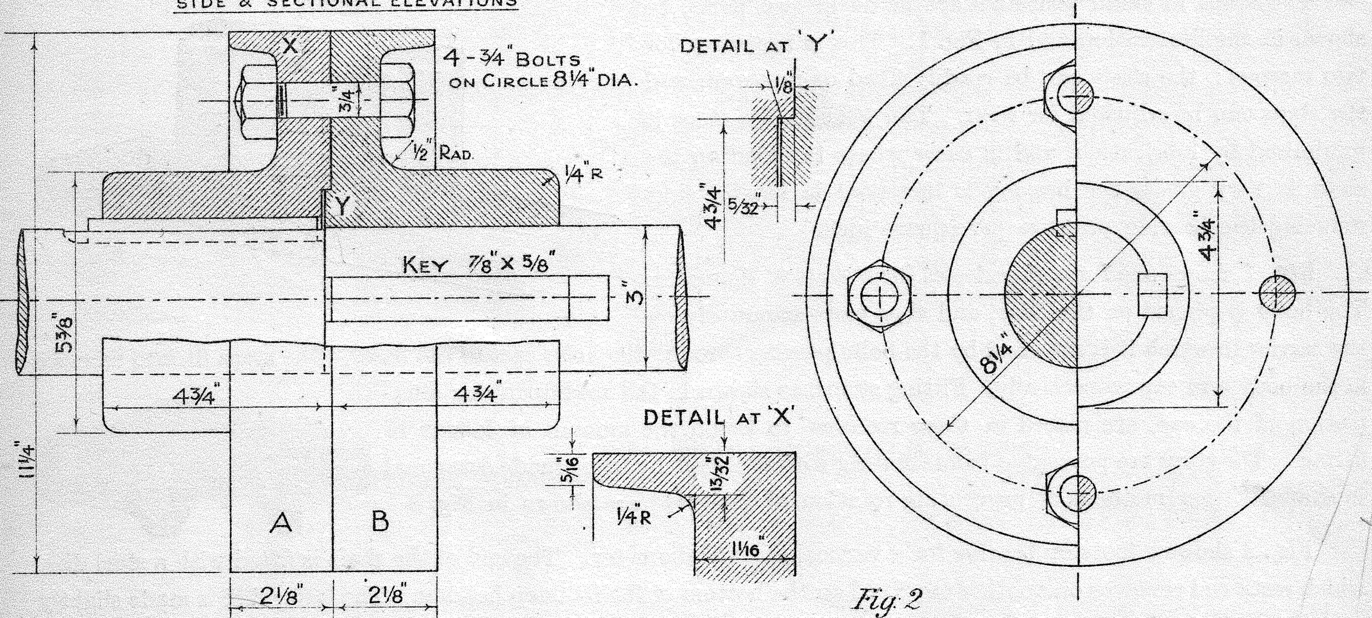


Fig: 2

CLAW COUPLING.

SIDE & SECTIONAL ELEVATIONS

SECTION AT A. B

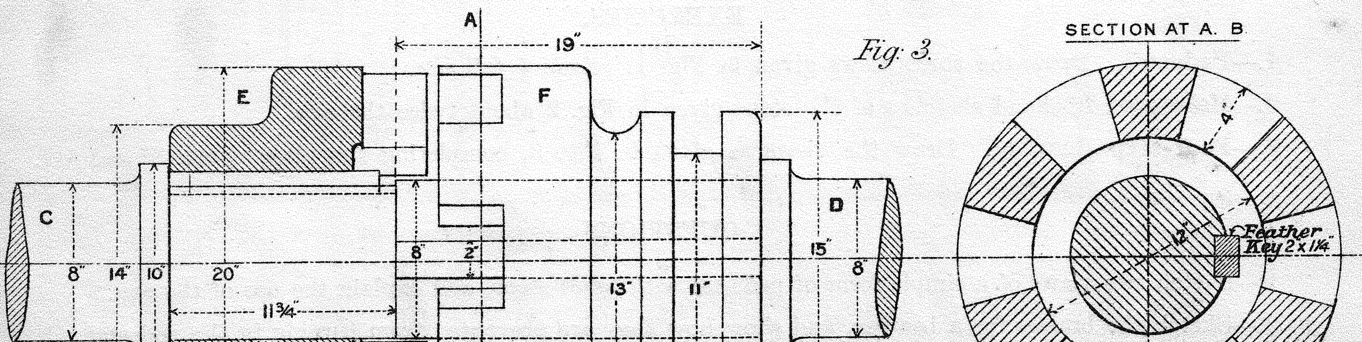


Fig: 3.

Scale 1/10 full size.

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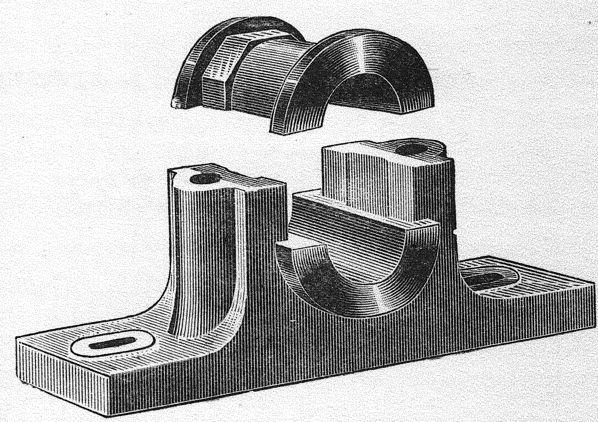
Plate XIV.—PEDESTAL AND FOOTSTEP BEARING.

A PEDESTAL is used for supporting a rotating shaft. The portion of the shaft which rests in the pedestal is called a journal. In a line of shafting it is usual to boss the shaft at the end through which the power is communicated The journal or neck is turned into this boss, leaving collars which fit against the faces of the pedestal brasses, and so the shaft is prevented from moving endways. A more economical method is to fit collars to the two ends of the shaft.

A pedestal consists of the following parts (see Fig. 1): (1) the cast-iron body C; (2) the gun metal steps or brasses E and F; (3) the cast-iron cap D; (4) the wrought-iron bolts for holding the cap and steps in position. The pedestal is carried in various ways—by a cast-iron wall box or bracket, as shown in Plate XV.; by a cast-iron standard, as shown in Plate XVIII., and in many other ways.



The steps in small pedestals are made in two parts, as shown in the illustration and in Fig. 1. This is necessary for two reasons: the shaft can be readily fixed or removed, and the steps can be adjusted for wear. The wear on the steps is minimised by lubrication, and in cases where the load on the shaft is very great, the bearing is increased in length, thus reducing the bearing pressure per square inch.

Fig. 1 shows one form of pedestal for a shaft 3″ diameter. The body is planed on the base, and thus rests accurately on the carrier to which it is attached by the bolts passing through the holes cast in the base. The steps fit into recesses in the body and cap respectively. Fitting strips, as shown in the accompanying illustration of the cap, are placed in these recesses to lessen the amount of labour in fitting. The steps are prevented from rotating with the shaft by being made octagonal in shape. Other methods of preventing rotation of the steps are shown in Fig. 2.

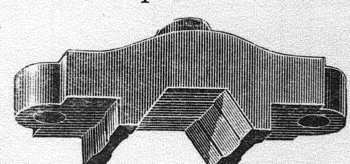


Fig. 3 shows a footstep bearing for a vertical shaft 3″ diameter. The end of the shaft is fitted with a steel disc, which rests and revolves on another one fixed on the bottom of the footstep bearing. The latter disc is made slightly conical on the underside to admit of a little adjustment, if the shaft end be not perfectly true.

The cast-iron body A is cored out and then bored to receive the brass bush B, which is turned to fit the hole at the top and bottom only.

EXERCISES.

1.—**Pedestal.** Draw the three views given in Fig. 1. *Scale $\frac{3}{4}$ full size.*

2.—Make good freehand sketches of the steps given in Fig. 2 about twice the size.

3.—**Foot-step Bearing.** Draw the views as given in Fig. 3, completing the plan. *Scale* 8″ = 1 *foot.*

QUESTIONS

1.—Sketch two views of a simple form of pedestal with brass steps, and explain the use of the cap.

2.—Sketch the brasses for a bearing, and show how they are prevented from turning in the pedestal.

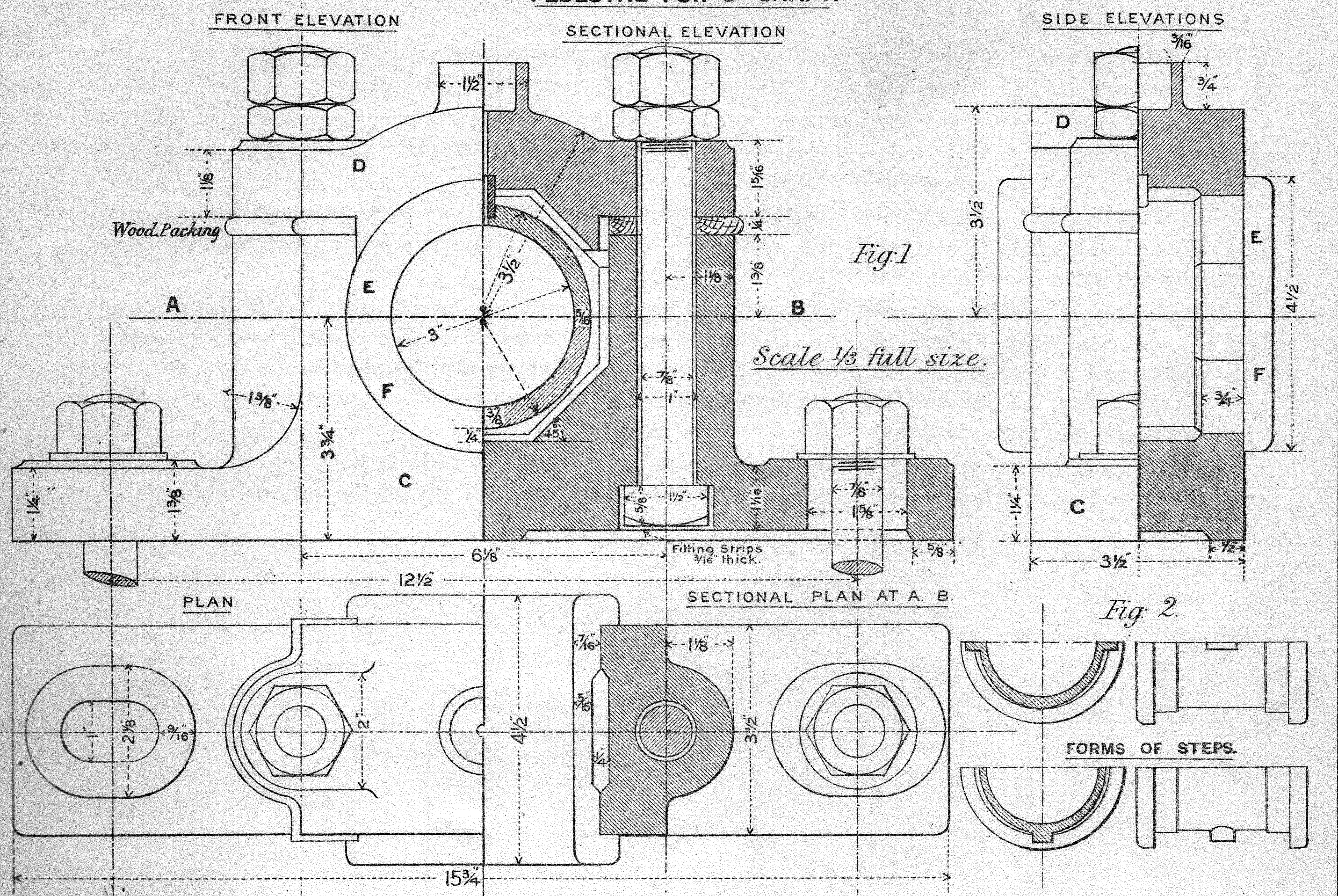
3.—Explain why a soft metal like brass is used for bearings.

4.—What are fitting or chipping strips? Give sketches showing where they are used, and explain their use.

5.—Why are the steps in the pedestal Fig. 1 separated horizontally?

SHAFT BEARINGS.

PEDESTAL FOR 3" SHAFT.



FOOT-STEP FOR VERTICAL SHAFT

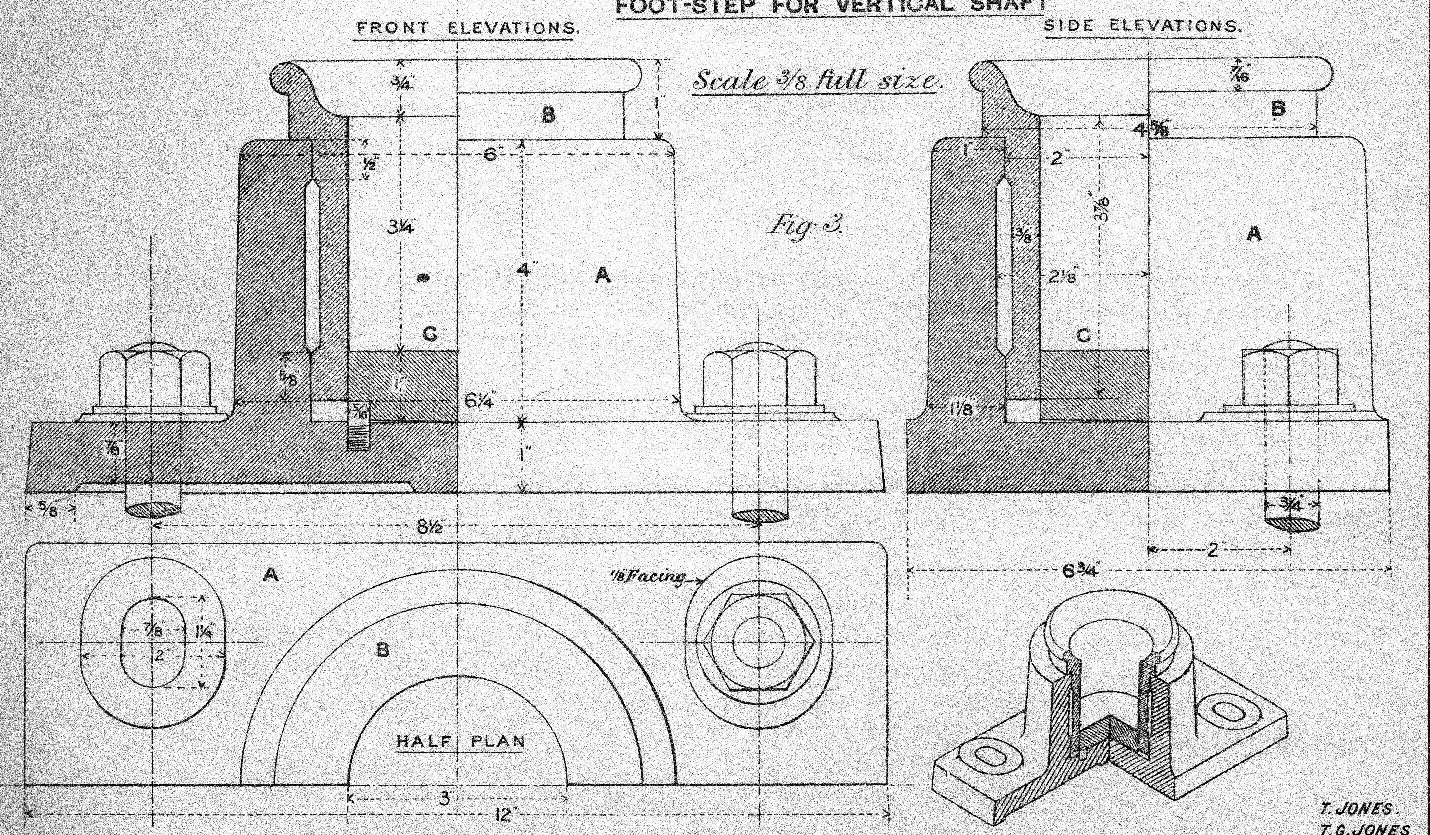


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Plate XV.—WALL BRACKET AND WALL BOX.

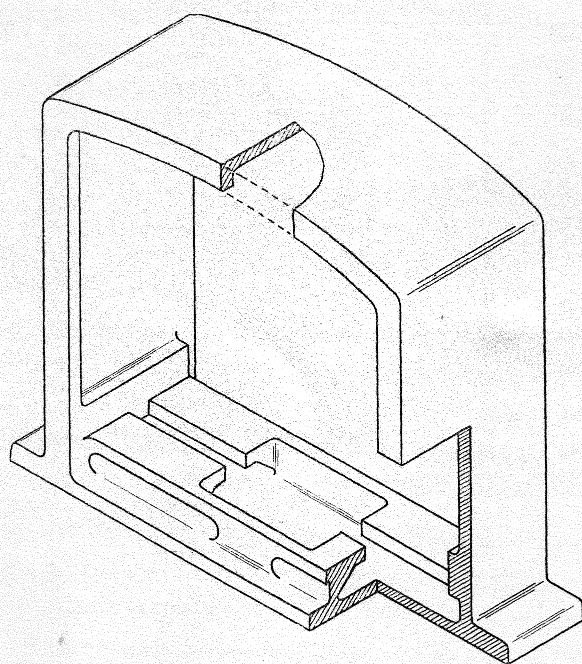
THE accompanying drawings represent two mill-gearing details—made by The Unbreakable Pulley and Millgearing Co. Ltd., of Manchester—for the supporting of shaft bearings or pedestals.

There are many types of pedestals, ranging from the rigid brass bearing plummer block—see Plate XIV.—to the various adjustable and non-adjustable swivel cast-iron bearings and roller bearings ; and any of these may be carried by the given side Wall Bracket or the Wall Box.

Side Wall Bracket.—This bracket is bolted to the wall by three $\frac{7}{8}''$ bolts, which pass through the wall, and at the other side the bolt heads fit into large cast-iron wall plates—thus distributing over a large area of the wall the pressure exerted by the bolts.

The pedestal is bolted to the machined horizontal seating, which runs parallel to the wall, and the supported shaft is therefore at right angles to the wall. Horizontal or side adjustment is made possible by the elongated holes in the seating and in the pedestal base, and with the non-adjustable type of pedestal vertical adjustment is secured by means of packing. If the shaft is to pass through the wall, only a minimum amount of cutting away of the wall is necessary with this type of support.

Wall Box.—This support, which is to carry a shaft through a wall, is built into the wall and does not entail the use of any holding down bolts. It is suitable for use with any of the various types of pedestals.



The accompanying isometric drawing assists one in realising its detailed construction. The pedestal is fixed to the machined seating S by two bolts fitted into the tee slots, and this arrangement permits of a considerable amount of horizontal adjustment. As with the Side Wall Bracket vertical adjustment of the shaft is also possible.

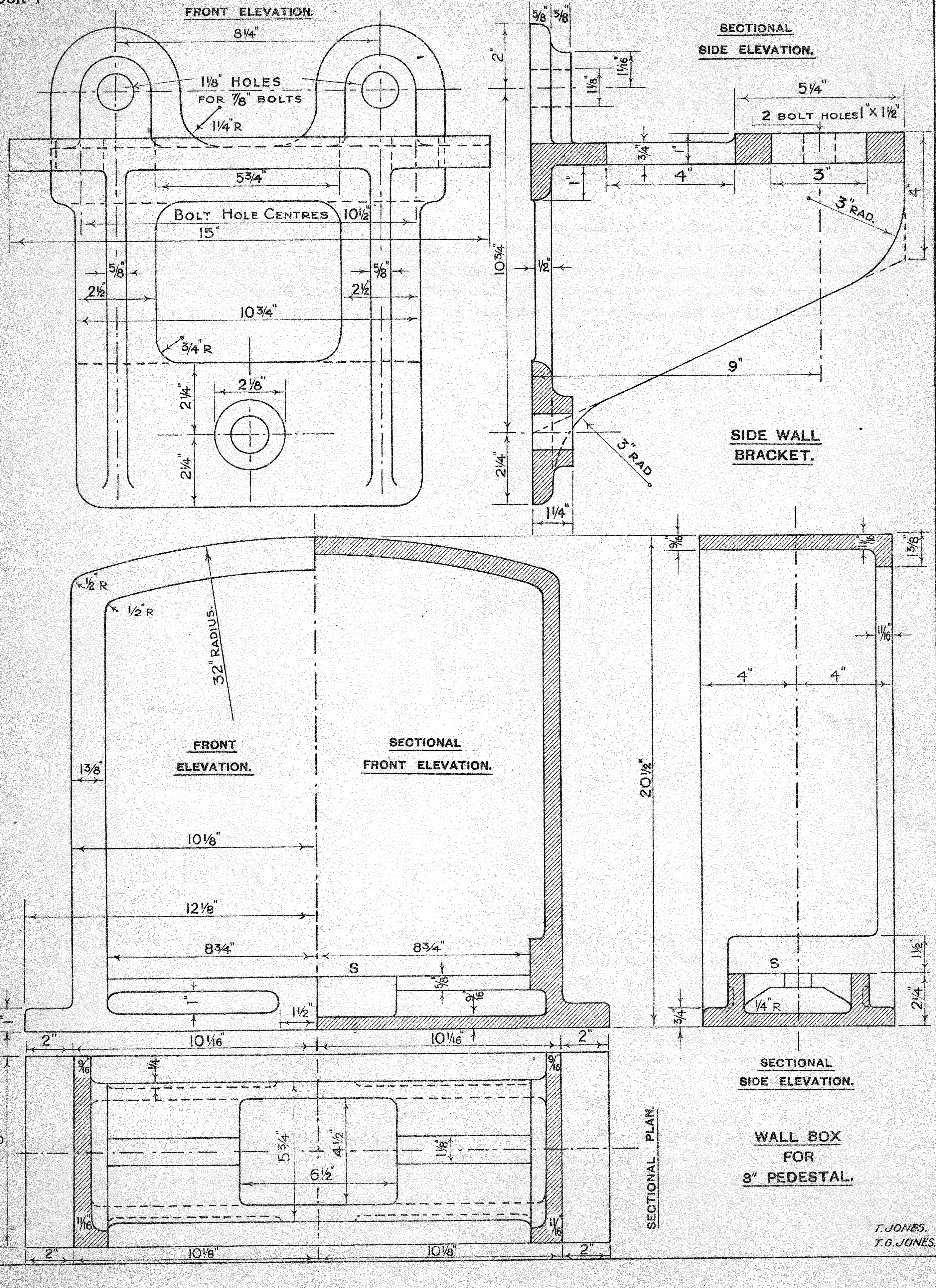
With this type of support the various forces exerted by the shaft on the bearing are most effectively taken up by the wall.

It will be seen from the drawings that the outside width of the box is less by $\frac{1}{8}''$ at the back than at the front, and the thickness of the metal of the vertical sides and the arched top varies uniformly from $\frac{11}{16}''$ at the front to $\frac{9}{16}''$ at the back face.

EXERCISES.

1.—**Side Wall Bracket.** Draw the given front and sectional side elevations ; and add the side elevation to the left of the front elevation, and the plan projected from the front elevation. *Scale half full size.*

2.—**Wall Box.** Draw the three given views, and add the back elevation to the right of the sectional side elevation. *Scale* $4\frac{1}{2}'' = 1$ foot.


FRONT ELEVATION.
1⅛" HOLES FOR ⅞" BOLTS
1¼" R
BOLT HOLE CENTRES
¾" R
SECTIONAL SIDE ELEVATION.
2 BOLT HOLES 1" × 1½"
3" RAD.
SIDE WALL BRACKET.
½" R
32" RADIUS.
FRONT ELEVATION.
SECTIONAL FRONT ELEVATION.
S
¼" R
SECTIONAL SIDE ELEVATION.
SECTIONAL PLAN.
WALL BOX FOR 3" PEDESTAL.
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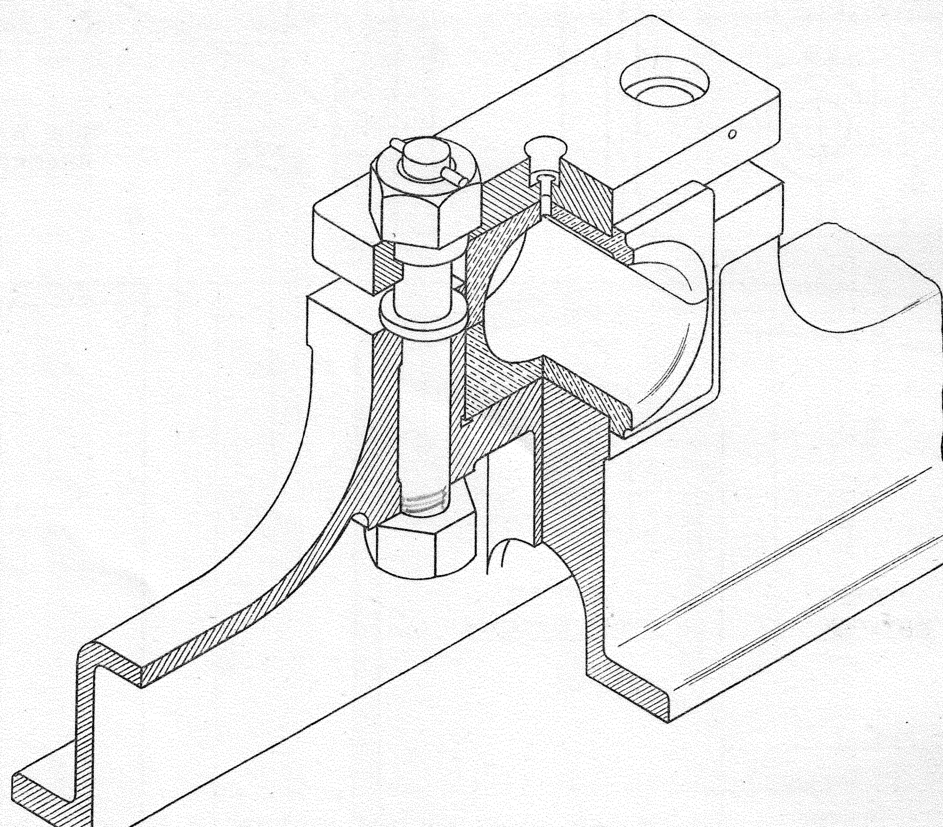
T. G. JONES.

Plate XVI.—SHAFT BEARING FOR VERTICAL ENGINE.

THERE are numerous designs of shaft bearings, but in the case of those for engine shafts the designs are less variable; and the accompanying detailed drawings and the isometric sketch represent a simple, rigid and efficient bearing for a small vertical engine.

The particular portion of the shaft supported is termed the **journal**, and the supporting detail, the **bearing**. The parts with which the journal is actually in contact must be of such bearing metal that it shall have sufficient strength to resist distortion when under load, have a high melting point and be able to resist abrasion without scoring the journal; these parts are called the **brasses.**

With perfect lubrication between the journal and the supporting surface there could not be any wear of either, but actually the brasses are of such a composition that they take practically all the wear resulting from imperfect lubrication, and must consequently be designed so that adjustment for wear may be readily made. For a small bearing the brasses are made in two parts, and the plane of separation, through the axis of the shaft, is at right angles to the mean direction of resultant pressure between the journal and the brasses—in this particular example the plane of separation is horizontal since the engine is a vertical one.



The top and bottom brasses are held rigidly in the bearing body—which in this case forms part of the engine bed—and are held together by the cap and the two cap bolts. It will be seen that with the intermediate collar on each bolt, the bolt remains firmly in position on the removal of the nut.

Provision is made for a syphon wick lubricator to be screwed into the bearing cap.

In the dimensioned drawing the several parts of the complete bearing are shown separately, but with the help of the isometric sketch above, which shows the parts assembled, there should be no difficulty in making a drawing of the complete bearing.

EXERCISE.

Draw the following views of the bearing, with all the parts assembled: (*a*) The front elevation, one half showing the central vertical section and the other the outside view; (*b*) the side elevation, one half showing the vertical section through the axis of the bearing and the other the outside view; (*c*) the plan—in projection with view (*a*)—one half showing the horizontal section through the axis of the bearing and the other the outside view. *Scale ½ full size.*

ENGINE BEARING FOR 4 INCH SHAFT.

(PARTS SHOWN SEPARATELY.)

Scale 3" = 1 Foot.

Note.—See the accompanying Isometric Drawing for the assembly of the parts.

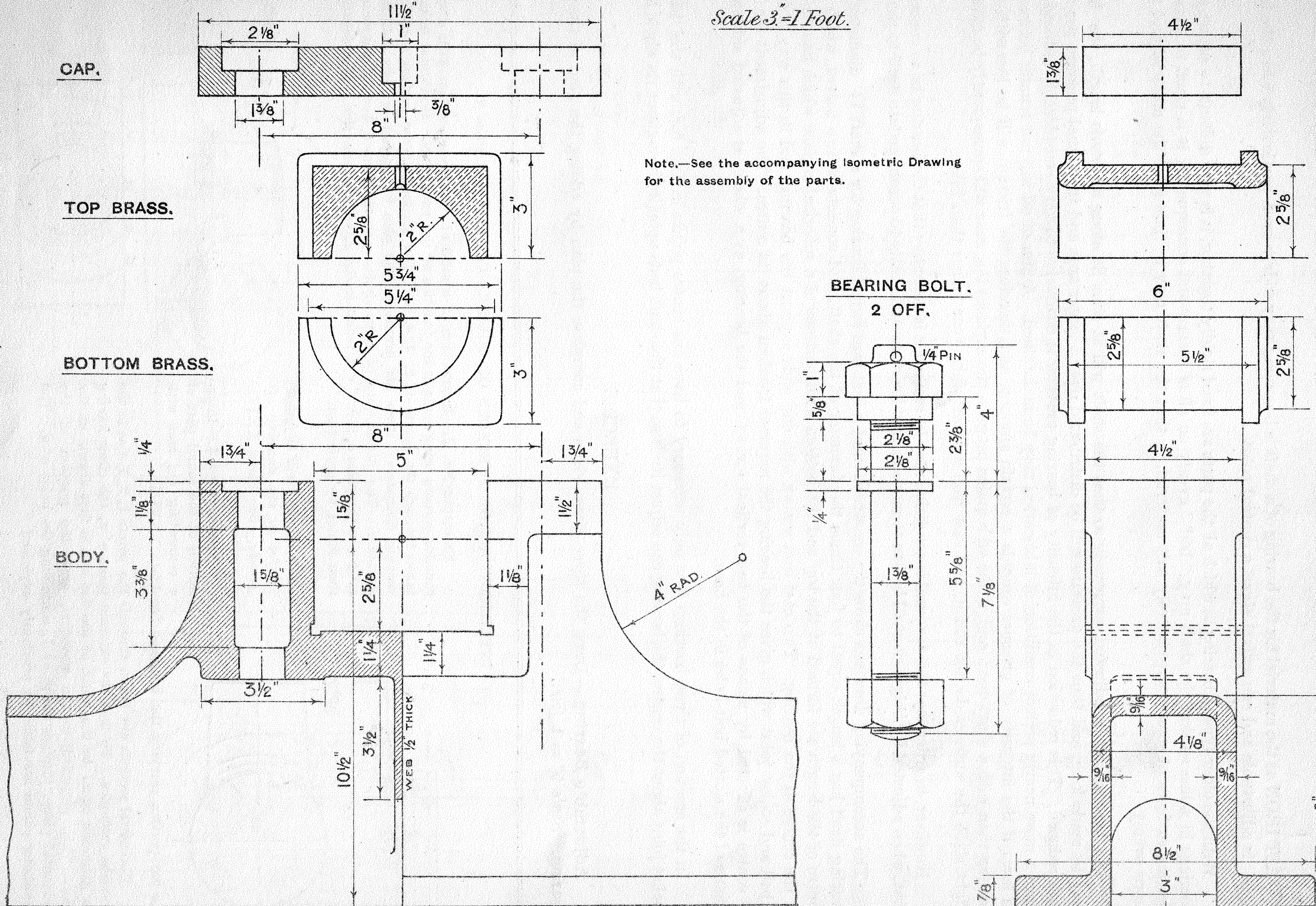


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Plate XVII.—ADJUSTABLE SWIVEL CAST-IRON BEARING.

CAST IRON as the material for the bearing of a shaft is becoming very widely used, particularly when the bearing is adjustable and can swivel about a central point.

With the older type of bearing in which all the parts are fixed rigidly together, it is very difficult to adjust several bearings in a line so that the shaft may " bed " evenly on all of them ; and consequently it is usual to have short bearing surfaces of some comparatively soft material, so that the shaft may in time wear the material and so bed more evenly.

If a bearing can be adjusted accurately so that the shaft will at once fit along its whole length, the bearing may be made long, and thus the pressure will be distributed over a larger area and the actual pressure per square inch reduced. This reduction in the intensity of bearing pressure is conducive to very considerably reduced wear, since a more nearly perfect condition of lubrication may be maintained. A film of oil will not remain between two su faces if the intensity of pressure exceeds a certain amount, and therefore the friction will be less with a long bearing than with a short one ; and, if the oil remains between the surfaces the wear will consequently be extremely slight since the surfaces do not touch, and cast iron will certainly wear longer than brass.

However, in order that this favourable condition may be secured it is necessary that the bearing be so constructed that, when adjusted for height, it shall be free to move in any direction sufficiently to accommodate itself to the shaft.

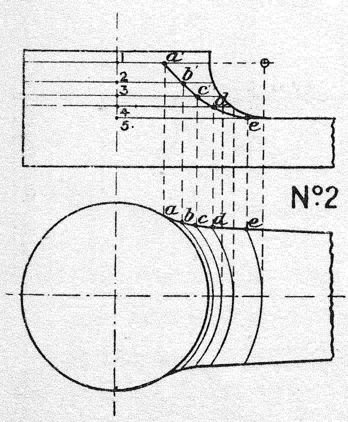
The accompanying drawings show the construction of a cast-iron swivel bearing for a shaft $1\frac{1}{2}''$ diameter. The bearing itself has a length equal to four times the diameter of the shaft, and consists of two halves, which in the centre of the length are formed at the top and the bottom with portions of a spherical surface which has its centre in the centre of the bearing. Two cast-iron screws with cup ends support the bearing, and in this way is formed a ball and socket joint which gives the bearing the required freedom about its centre. The adjustment for height of centre is effected by means of the two vertical screws, and the bearing, as a whole, is adjusted sideways by means of the slotted bolt-holes in the base.

In the erection of these bearings it is only necessary to have the centre points in line, for, with the ball and socket joint about the centre, the bearing can adjust itself so that the shaft beds upon it throughout its whole length.

EXERCISE.

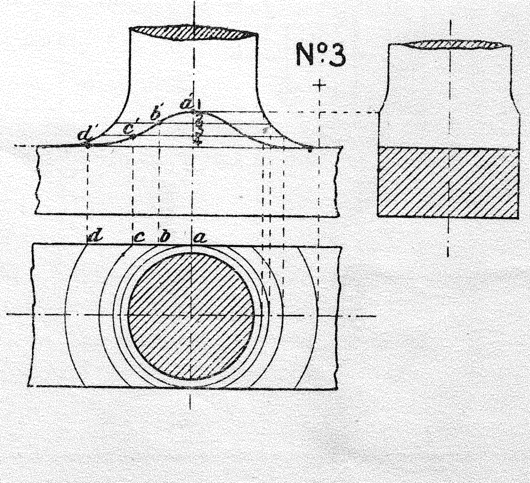
Adjustable Bearing.—Draw the three given views and complete the plan by adding the plan of the swivel bearing. *Scale* 9″ = 1 *foot.*

INTERSECTIONS OF SURFACES OF SOLIDS.—*Continued from Plate X.*



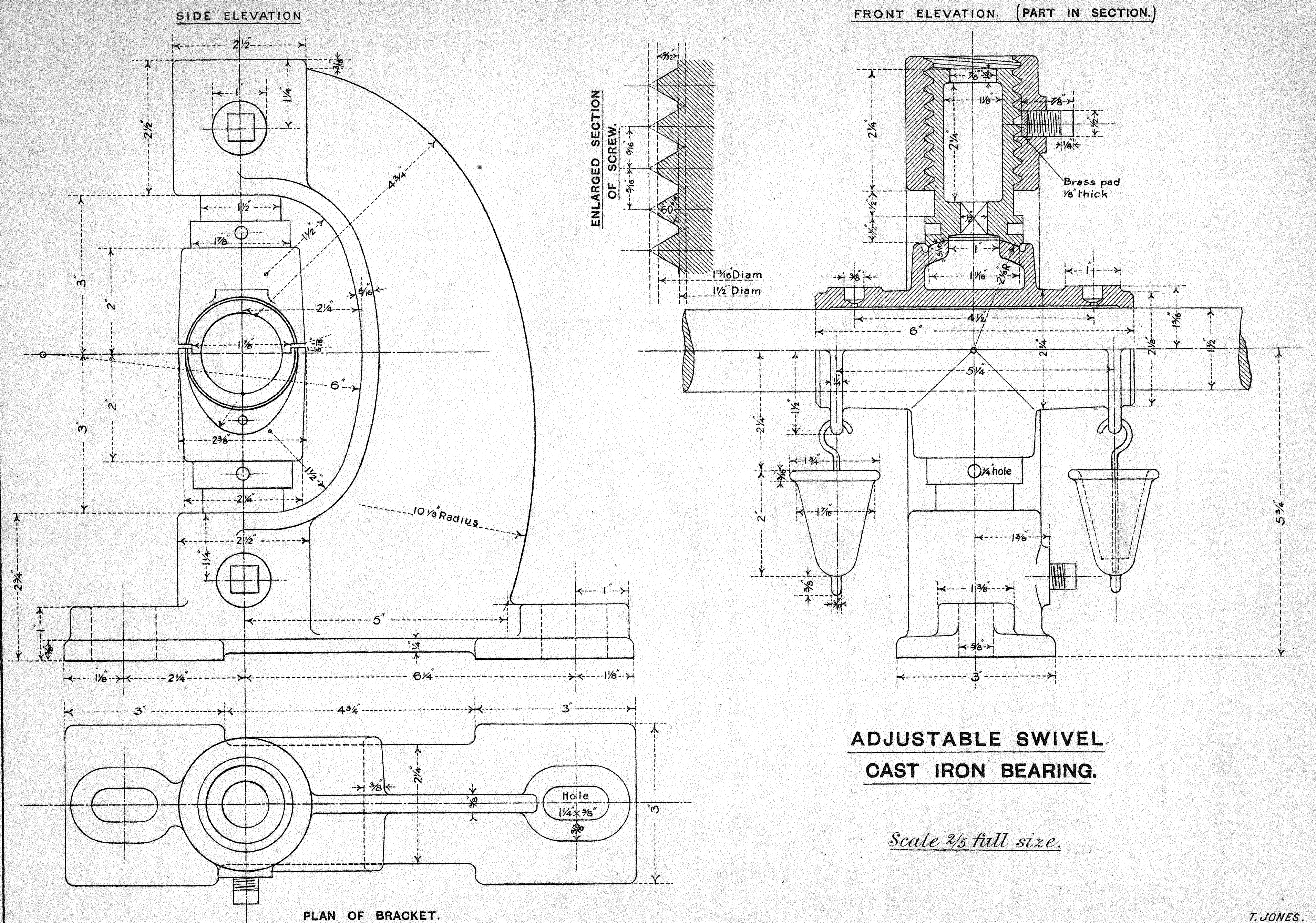
Example No. 2.—Intersection of a solid of rectangular section with a cylinder, rounded corners being made at the junction of the two solids. Horizontal sections are taken through the points 1, 2, 3, 4, 5. The sections of the surface are circles of varying diameters: these are drawn in the plan by projecting their radii from the elevation. The intersections of these circles with the vertical faces of the rectangular solid give the plans *a*, *b*, *c*, *d*, *e* of points in the curve of intersection. The elevations are then found by projection.

This example will be found useful in the drawing of cranks.—See Plate XXIV.



Example No. 3.—Intersection of the round part of a connecting rod end with the flat portion carrying the brasses.—See Plate XXVIII.

Horizontal sections are taken through the points 1, 2, 3, 4 as in the previous examples. To find the extreme point a' a, the circle having its diameter equal to the width of the flat end is first drawn in the plan. This, projected to the elevation, gives the position of the hori zontal plane, through the point 1, containing the point a' a. The remaining points are found as previously explained.



ADJUSTABLE SWIVEL CAST IRON BEARING.

Scale 2/5 full size.

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Plate XVIII.—BEARING AND STANDARD FOR SHAFTS.

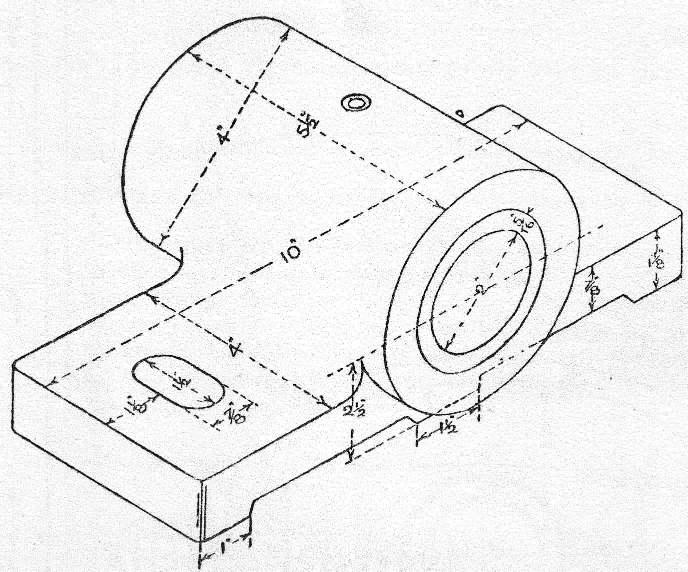
FIG. 1 gives two views of a cast-iron bearing for the crank shaft of a small pumping engine for a torpedo boat The bearing is supported by two steel pillars, and is held in position by steel cotters. The steel pillars are held to the base plate of the engine by nuts screwed on the ends. The inverted cylinders are attached to the top end of the pillars in the same manner. It may be noted that the cap is held down by stud bolts, the central web preventing the use of ordinary bolts.

Fig. 2. This standard is used to carry a 12″ pedestal, for the first motion shaft of a large double acting pump, driven by a water wheel. It is made of box section, and is a good example of a large hollow casting. The flat sides are strengthened by the brackets shown in the sectional elevation. The pedestal is held in position by T headed bolts, which are passed down through the rectangular holes shown in the plan. The standard is secured to the bed of the pump by six bolts.

EXERCISES.

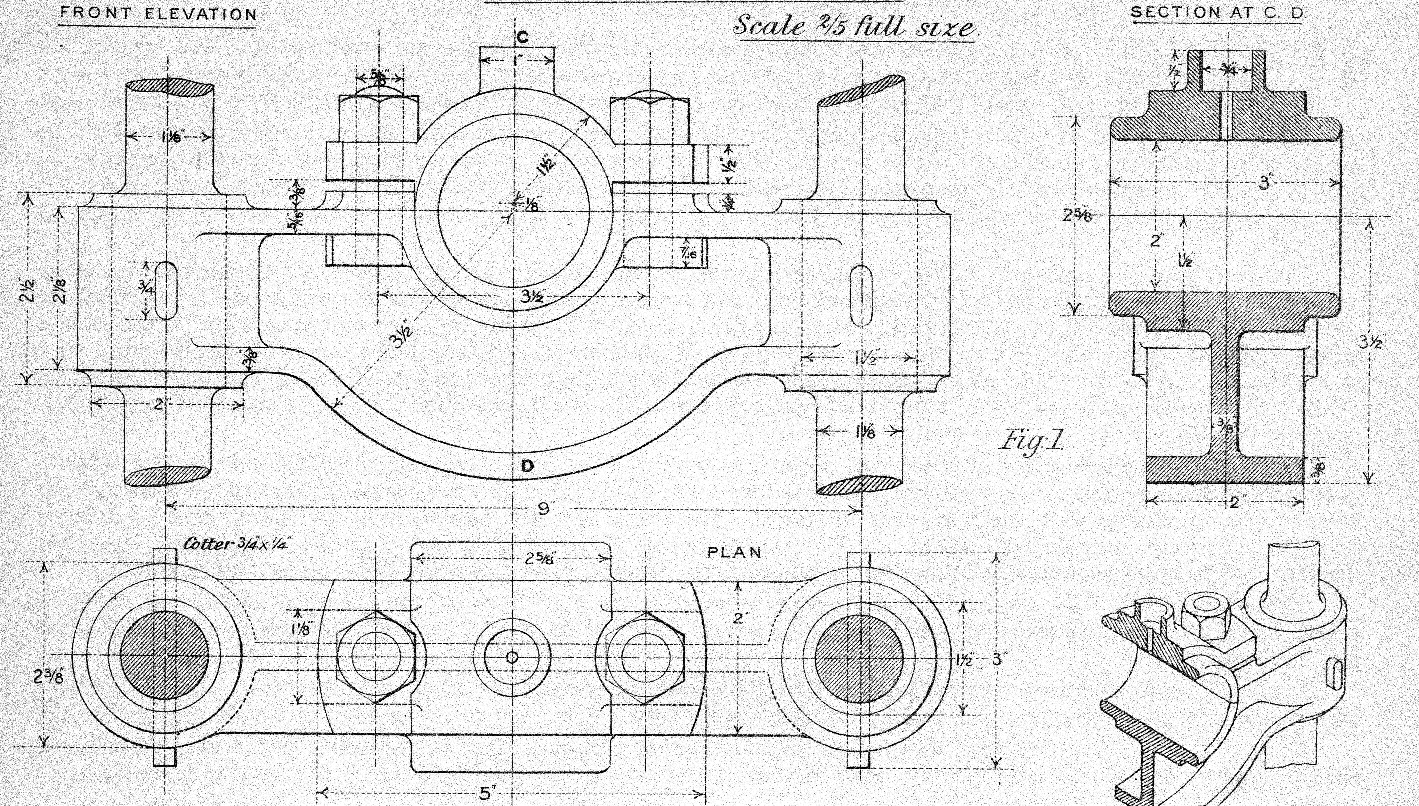
1.—Cast-iron Bearing. Draw the three views as given, and add an end elevation. *Scale full size.*

2.—Standard. Draw the standard as given, and add an end elevation. *Scale ⅛ full size.*



3.—Draw front and end elevations, and also a plan of the *brass bushed* cast-iron bearing, shown in the accompanying isometric sketch. *Scale 9″ = 1 foot.*

CAST IRON BEARING FOR SHAFT.



CAST IRON STANDARD FOR PEDESTAL.

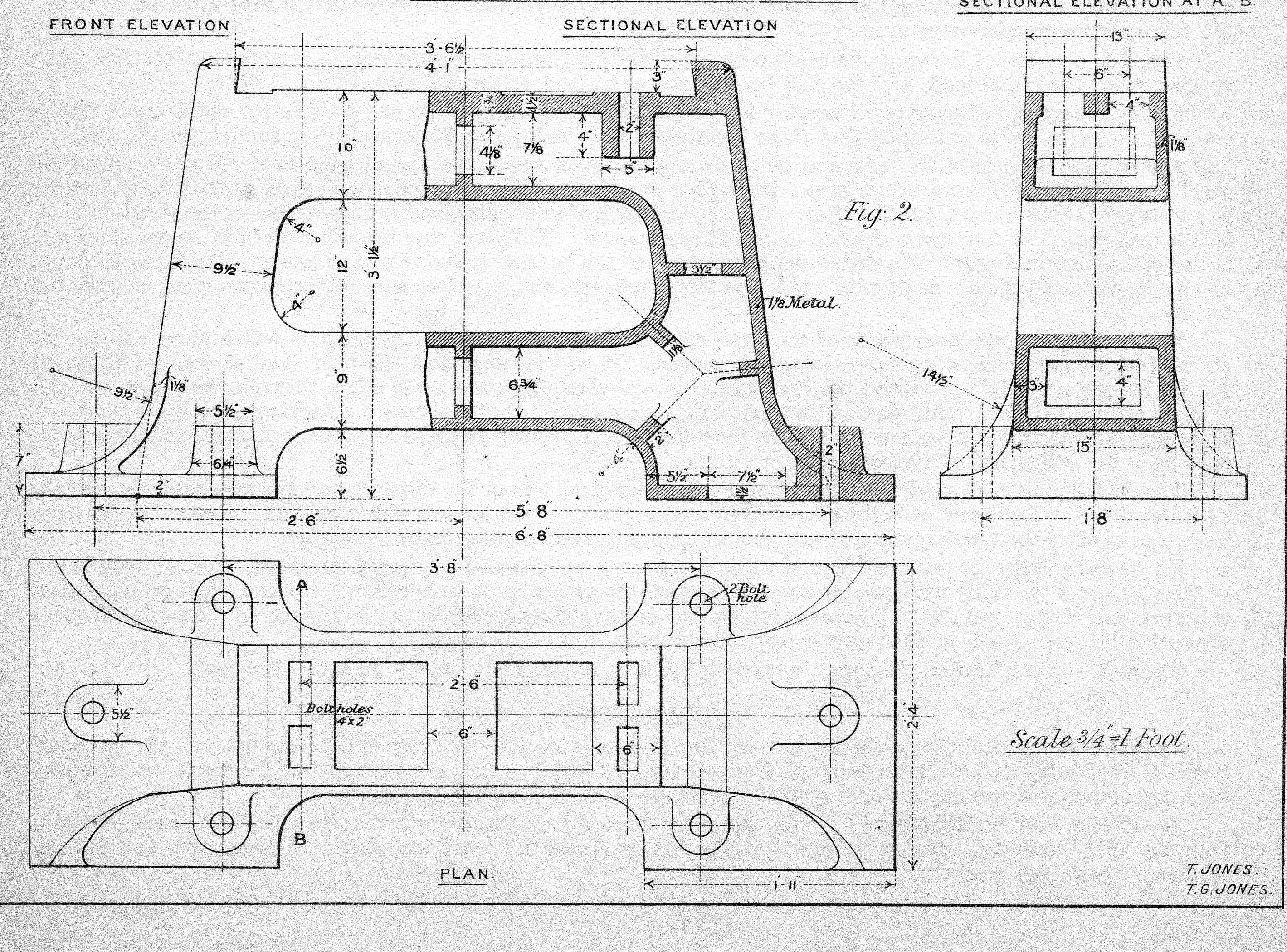


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Plate XIX.—BALL AND ROLLER BEARINGS.

BALL BEARING. Fig 1 represents a sectional view of the Skefko self-aligning double-row ball bearing. The complete bearing consists of an inner ring R_1, an outer ring R_2—both of special quality steel—and between them two rows of hard steel balls, which are retained in their correct positions by a thin metal cage.

Rings. The *inner ring* is a light driving fit on the shaft, and is pressed against a shoulder on the shaft by means of a circular nut locked by a grub screw. This ring is provided with two races, one for each row of balls, and they are so designed that the tangents of the balls at their points of contact with the inner and outer races are parallel, and there is thus no tendency for the two rows of balls to be forced together causing an axial pressure on the cage.

The *outer ring* is a piston fit in its housing and free to move laterally: by this means the ring is able to rotate very slowly and so equalise the wear or distortion of the outer race. The surface of the outer race is spherical, its centre being the centre of the bearing, thus allowing the balls, together with the cage and inner ring, to move as a whole within this race; in this way the bearing is capable of adjusting itself to any deflection of the shaft upon which it is mounted. Also, it will be seen that the lines drawn through the contact points of all balls meet at the centre of the shaft, and thus the surface of rotation of each set of balls is conical, providing for the resistance of axial thrust in either direction.

The **Cage** is a single piece of sheet-iron pressed to shape. The cage itself cannot hold the balls, but when it is assembled with the inner ring small chambers are formed in which the balls are placed and kept in position without in any way interfering with their freedom to rotate. The small bent tongues between the balls serve to prevent those in either row touching one another. The appearance of the cage is indicated by the sketch, Fig. 3, on the drawing. (Dimensions of this detail are not given, and the student must represent it to the best of his ability.)

The bearing is totally enclosed by the covers secured to the two faces of the housing. The cover through which the shaft passes is provided with a circular groove in which is placed an oiled felt washer to exclude dust and moisture.

Such a bearing requires very little attention. The lubricant must be chemically neutral; and for general purposes a mixture of vaseline and vaseline oil is recommended. For high speeds a good mineral oil is preferable.

Load. If radial bearings are subjected to an axial load at the same time as the radial load it can be reckoned that the radial load plus three times the axial load must not exceed the total load which the bearing is designed to carry.

The maximum steady load for the bearing illustrated is about 2,600 lb. at 300 r.p.m. to 1,500 lb. at 1,500 r.p.m. at ordinary temperatures (up to 120° F.). If these temperatures are exceeded the load must be reduced; the temperature should never exceed 390° F.

Fig. 2 is a sectional drawing of a Hoffmann roller and ball bearing suitable for an electric motor. The roller bearing takes the radial load, and the ball bearing the axial load or thrust.

Roller Bearing. This type of bearing is not intended to supersede the ball bearing for radial loads, but is employed when space is so limited that there is no room for a ball journal bearing strong enough for the load.

The bearing consists of an inner and an outer ring, between which is a row of hard steel rollers in a retaining cage. The inner ring is grooved to form a track for the rollers, while the outer race is plain so that the rollers are free to take up their correct position in it. The cage is made of gun-metal, and is represented in the sketch, Fig. 4, on the drawing. *The diameter and length of the rollers are equal.* The inner ring is made a tight fit on the shaft and is clamped tightly endways. The outer ring is a push fit in the housing and also held endways. The bearing cannot be used to take end thrust, or even to locate the shaft endways, and therefore some other means must be provided for this.

The **Double-thrust bearing** is of the type provided complete with sleeve and nut which gives adjustment of the bearing independently of the clamping pressure. It will be seen that the mild steel sleeve—which fits on the shaft—projects slightly beyond the nut, and thus any clamping pressure is taken through the sleeve and not through the balls. As an additional precaution that this condition is being obtained a packing piece is used between the roller bearing and the flanged sleeve, the face of which is cleared away so as to be in contact with the flange only over the solid part of the sleeve.

The central hardened steel washer is clamped between shoulders in the housing, and the two outer ones rotate with the sleeve. Each row of balls is held in a retaining cage of gun-metal, which prevents friction between the balls, and enables the bearing when dismantled to be handled without the balls falling out.

The bearing is totally enclosed, and the inner end cover is thickened up round the shaft, bored to ·004″ clear of the shaft, and provided with four grooves into which the grease finds its way, and makes a more or less perfect seal against moisture and dirt. When assembling the housing should be filled with grease, and a Stauffer or other form of lubricator fitted so that grease may occasionally be forced in.

The safe working load on the thrust washers is 1,800 lb. at 100 r.p.m. to 400 lb. at 2,000 r.p.m.

EXERCISES.

1.—**Ball Bearing.** Draw the *given view*, Fig. 1, and add the *end elevation*—to the left of the sectional elevation—with the dished cover removed, the *end elevation* looking on the broken end of the shaft, and the *plan* with the covers and housing only in section. *Scale full size.*

2.—**Roller and Ball Bearing.** Draw the *given view*, Fig. 2, the *end elevation* to the right of the section—with the cover removed—the *end elevation* to the left of the section, and the *plan* with the covers and housing removed. *Scale full size.*

BALL AND ROLLER BEARINGS.

SKEFKO BALL BEARING.

SECTIONAL ELEVATION

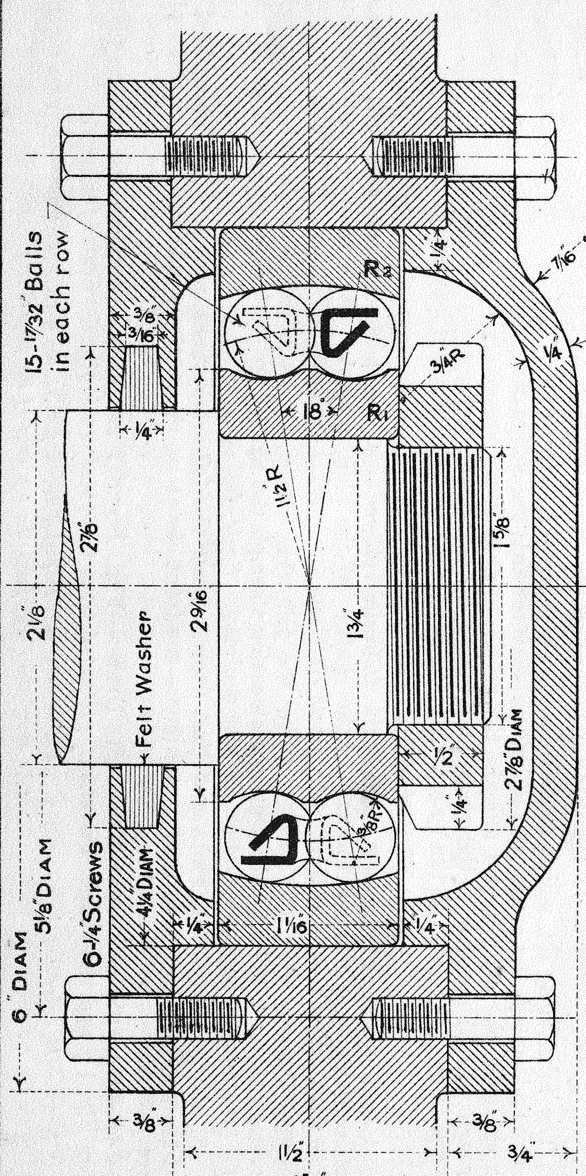

Fig. 1.

HOFFMANN ROLLER JOURNAL AND THRUST BEARINGS.

SECTIONAL ELEVATION

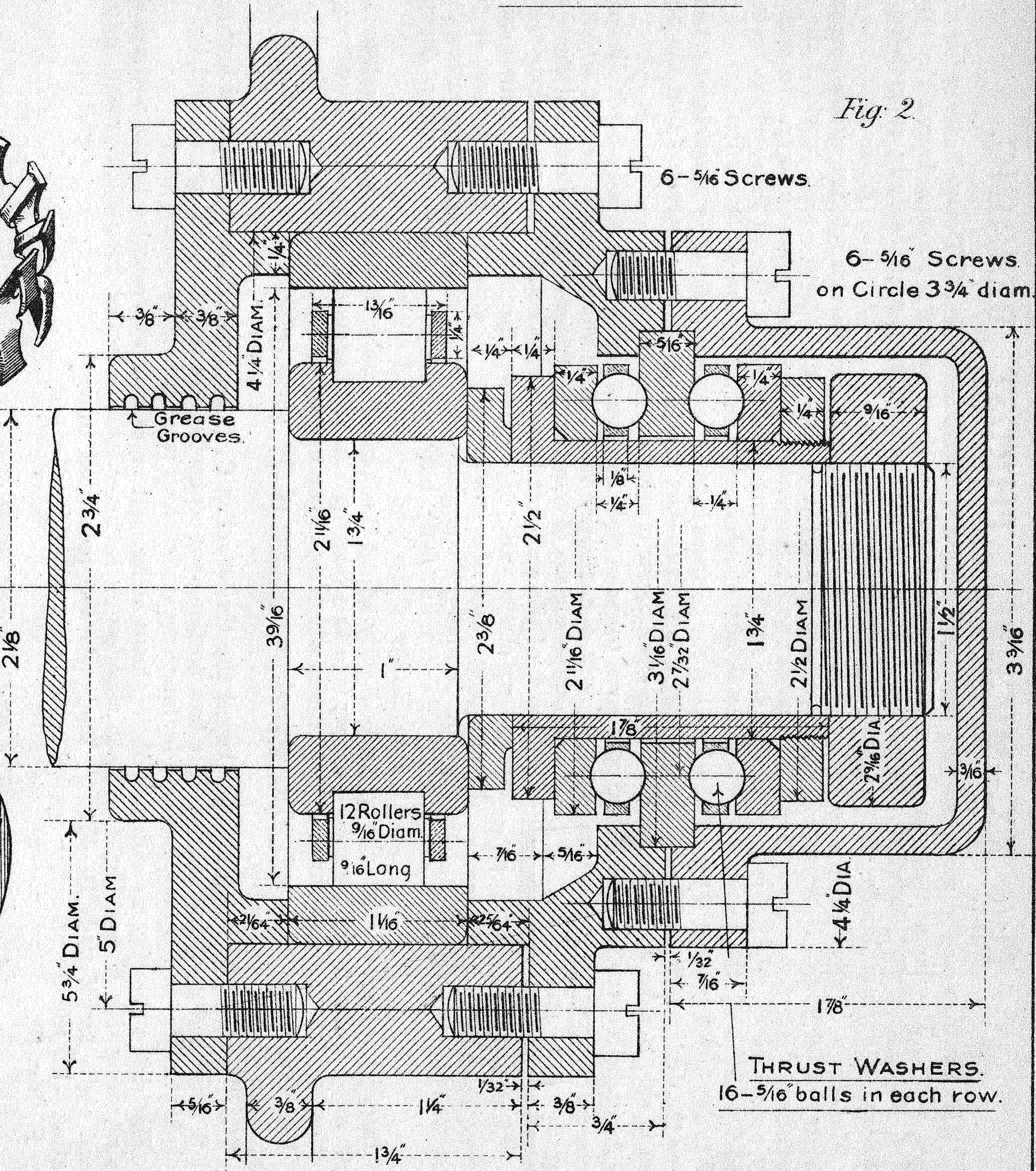

Fig. 2.

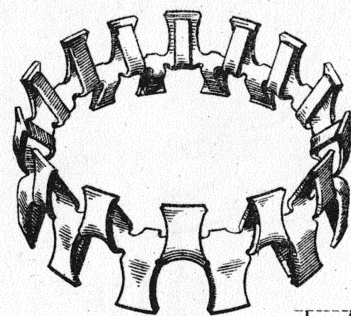

Fig. 3.

Figs. 3 and 4 are for reference only, not to be drawn. See the text.

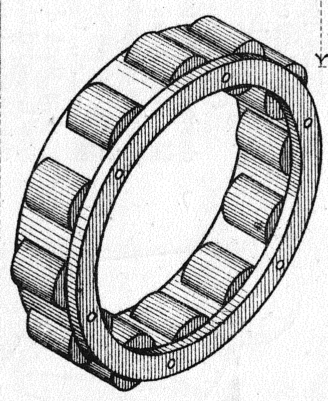

Fig. 4.

Scale 7/8 full size.

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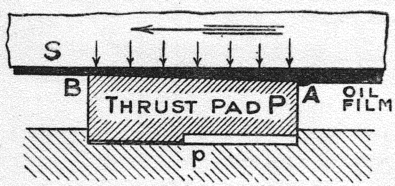
Plate XX.—MICHELL THRUST BEARING.

THE drawings on the accompanying plate represent the details of a Michell Thrust Bearing for a horizontal shaft. The design embodies a new idea which is based on the theory of lubrication founded by Prof. Osborne Reynolds; and therefore, before the striking features of this bearing can be fully appreciated, it will be necessary to explain briefly the **underlying principles.**

The object of lubrication is to keep the two surfaces, between which there is relative motion, separated by a thin film of oil in which pressure is automatically generated to balance the load; and, provided the oil film be maintained the surfaces never come into contact and consequently there cannot be any wear. In such a case the only resistance to relative motion is that due to the continual distortion of the oil film—which varies with the viscosity of the oil, the speed of relative motion of the surfaces and the thickness of the oil film.

It can be shown that if the oil film between the surfaces is to be maintained *the two surfaces must not be parallel*—hence the design of the ordinary thrust bearing is faulty,—but they must be so constructed that the film between them is able to take up a tapered formation when under pressure.

A simple illustration, as follows, may serve to make clear the point:—Consider a body S with its oiled surface moving over a stationary rectangular thrust pad P against which it presses. The oil on the surface of S enters the space between itself and the thrust pad at the edge A of the pad, but leaves the effective film space along the remaining three edges of the pad. If, therefore, the film is to be maintained effectively the opening along the leading edge A must be greater than at the leaving edge B. Actually, the angle of inclination of pad to moving surface is extremely small (about 1 in 3,000), but it is subject to slight variation depending upon the conditions of the load, speed and oil viscosity.

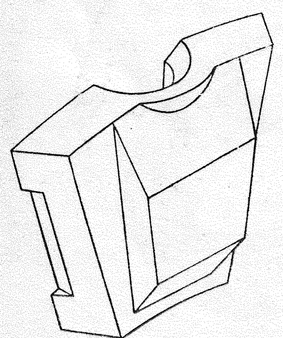


In order, therefore, that the pad may readily adjust itself to the exact angle which the conditions require, it is cut away at the back, from the entering face to a line "p" (a little beyond the geometrical centre), called the pivoting line, which passes through the point at which acts the resultant pressure of the oil film.

Description of Bearing. The general arrangement of the parts is indicated in Figs. 1 and 2, and some details are shown separately.

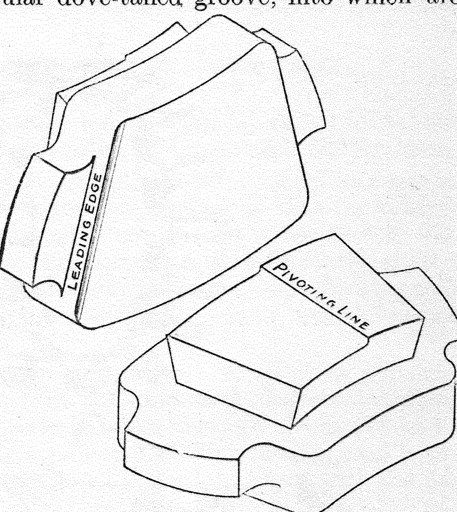
The thrust shaft, upon which is keyed the thrust collar C, is surrounded by the top and bottom halves, A and B respectively, of the casing. The two portions are connected together by four screws, and the casing as a whole is attached to the machine body by four long studs which pass through its whole length.

Placed in the casing on each side of the thrust collar, and prevented from rotating by the vertical screw stop, is a retaining ring E (see Fig. 4). Each ring has on its inner face a circular dove-tailed groove, into which are placed eight thrust pads D—Fig. 5 The pads are prevented from rotating with the thrust collar, against which they press, by the pad stop F (Fig. 6).



(The details D and F are also indicated in the accompanying isometric drawings.)

One set of thrust pads must be designed to take the thrust of the collar for clockwise, and the other set for anti-clockwise rotation. It will be noticed that the mean radius of the pads is 57·75 mm.—i.e., the resultant pressure on each pad is that distance from the axis of the shaft,—and the pads touch adjacent ones on this mean circle through the curved end lugs.



The lubricating oil, which must be perfectly filtered if wear is to be avoided, enters the casing at the bottom, rises towards the centre of the shaft by the oil passages outside the retaining rings, then passes through the oil grooves in the rings, and thence by centrifugal force outwards over the faces of the collar. The oil is allowed free egress after leaving the collar.

EXERCISES. (NOTE: All dimensions are in millimetres.)

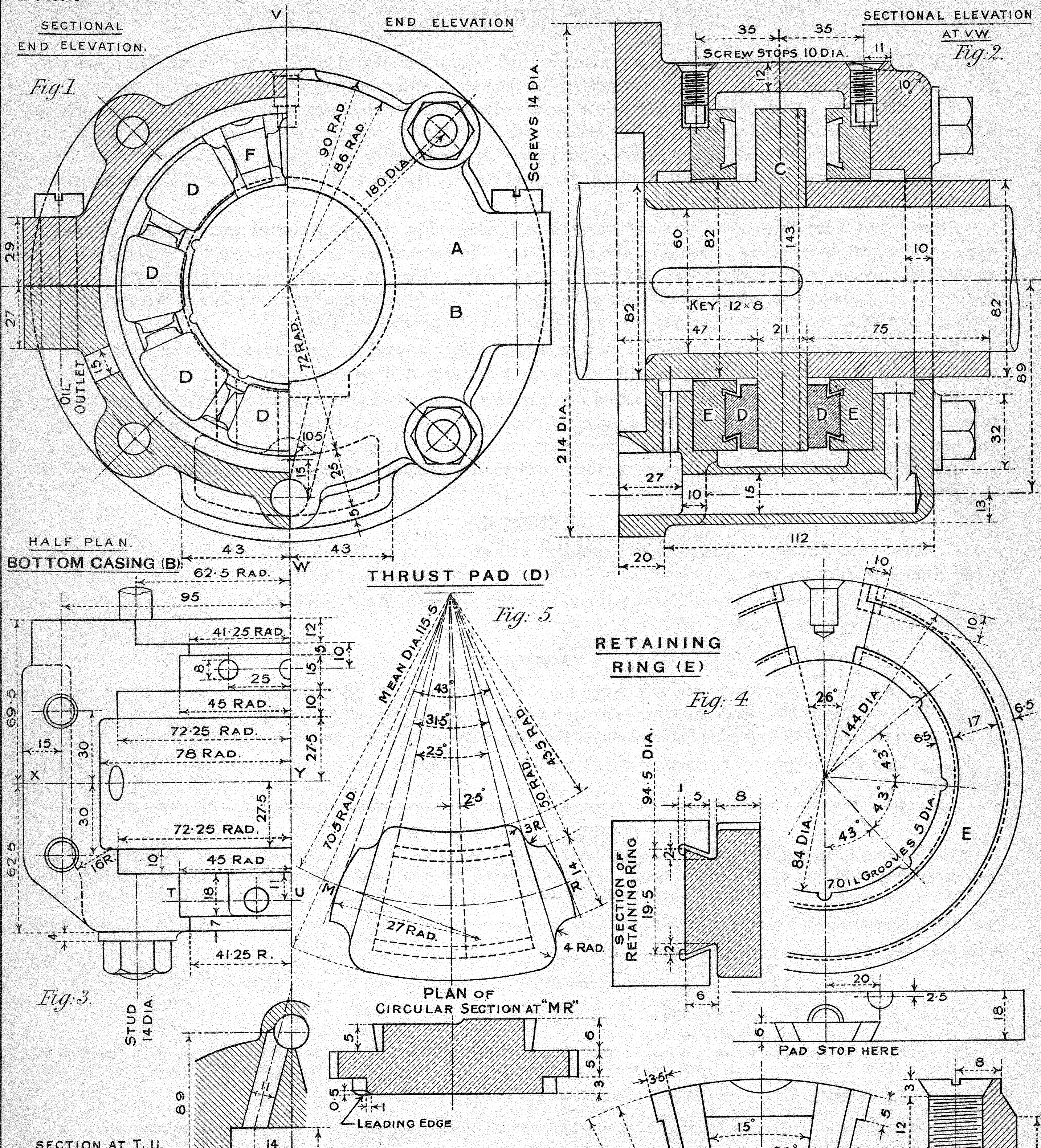
1.—**Top Half of Casing.** Draw the following views:—(*a*) the half sectional and end elevation; (*b*) the sectional elevation through the axis of the shaft; (*c*) the plan, projected from view (*a*); (*d*) the side elevation projected from the plan to the right. *Scale, full size.*

2.—**Bottom Half of Casing.** Draw the following views:—(*a*) the half end elevation and half sectional elevation through TU; (*b*) the sectional elevation through the axis of the shaft; (*c*) the plan, projected from view (*a*); (*d*) the side elevation, projected from the plan. *Scale, full size.*

3.—**Retaining Ring, Pads and Pad Stop.** Draw the following views, with the pads assembled in the ring and the pad stop on the vertical centre line:—the front elevation, the sectional side elevation and the plan. *Scale, twice full size.*

4*.—**Complete Thrust Bearing.** Draw:—Figs. 1 and 2 as given; the complete outside plan, projected from Fig. 1; and the outside elevation, projected from the plan to the right. *Scale, full size.*

* NOTE.—*In order that the views may be drawn on a half Imperial drawing sheet, the horizontal centre-line for the elevations must not be more than* $4\frac{3}{8}''$ *from the top edge of the paper.*



MICHELL THRUST BEARING

NOTE: ALL DIMENSIONS ARE IN MILLIMETRES

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Plate XXI.—CAST-IRON BELT PULLEYS.

PULLEYS are used for transmitting motion from a shaft to another one which is parallel to it. The connection is made either by belts or ropes. The material of the belt is either leather or a special woven canvas; and that of the rope, cotton or hemp. The belt is made endless, and stretched tightly over the pulleys; the driving being effected by the friction between the belt and the rim of the pulley. A pulley consists of three principal parts: the rim, the arms and the boss, which are cast in one piece. By means of the boss the pulley is secured to the shaft. The arms, which are made taper, radiate from the boss and connect the rim to it The width of the arms at the rim is two-thirds the width at the boss.

Figs. 1 and 2 are drawings of a pair of cast-iron belt pulleys, Fig. 1 showing curved arms, and Fig. 2, straight arms. The arms are elliptical in section; the axes of the ellipse are usually in the ratio of 2 : 1. Fig. 3 shows a method of drawing approximately this ellipse by arcs of circles. The rim is made convex in form, the radius of the curve being about 3 w, where w = width of the pulley. This form of rim keeps the belt in the centre, since every portion of it tends to move to the greatest diameter of the pulley.

Fig. 4 gives an example of a cast-iron cone or speed pulley, as used for driving machines or tools, where a *variable* speed is required to be transmitted from a shaft running at a *constant* speed.

The velocity ratio of shafts, driven by pulleys, is inversely proportional to the diameters of the pulleys keyed on them, assuming no slip to occur. Suppose a pulley n'' diameter keyed to a shaft making A revolutions per minute; and a pulley m'' diameter keyed to a shaft making B revolutions per minute; then $n : B :: m : A \therefore n A = m B$. $\therefore$ it follows that the diameter of pulley $\times$ revolutions of shaft is constant for any pair of shafts connected by belt and pulleys.

EXERCISES.

1.—**Cast-iron Pulleys.** Draw the two cast-iron pulleys as given in Figs. 1 and 2. *Scale* 3″ = 1 *foot.* Show a full sized section of an arm.

2.—**Cone Pulley.** Draw the sectional and end elevations given in Fig. 4, adding a plan and an end elevation looking inside the pulley. *Scale* ½ *full size.*

QUESTIONS.

1.—Determine the maximum and minimum velocities of the cone pulley, supposing it to be driven from a countershaft running at 100 revolutions per minute, by a pulley of the same dimensions.

2.—Determine also the variable horse power of the belt, which is $2\frac{1}{4}''$ wide, under the same conditions.

3.—Taking the pulley Fig. 1, running at 150 revolutions per minute, find the horse power of the belt, which is 7″ wide.

HORSE POWER OF LEATHER BELTS.

When a belt is in motion the tension in one side is greater than in the other. The power which can be transmitted depends upon the velocity of the belt and the *effective* tension, which latter is the *difference* between the tensions in the tight and slack sides. The ratio of these tensions varies with the arc embraced by the belt, and the co-efficient of friction between the belt and the pulley. Prof. Unwin gives a table of these ratios $\frac{T_1}{T}$, from which the following values are taken; co-efficient of friction, = ·4. T_1 = tension in the tight side; T = tension in the slack side :—

(1) $\frac{T_1}{T} = 2{\cdot}4 = \frac{24}{10}$ for an arc of 126° – pulleys 10′ and 1′ – 10′ centres.
(2) ,, $= 3 = \frac{30}{10}$,, ,, 156° – ,, 5′ and 1′ – 10′ ,,
(3) ,, $= 3{\cdot}5 = \frac{35}{10}$,, ,, 180° – ,, 4′ and 4′ – 10′ ,,

The maximum safe working stress in a leather belt, through the laced joint, is 360 lb. per square inch = 90 lb. per inch of width for a belt ¼″ thick. If in each of the above examples a belt 10″ wide be assumed, the total safe working stress = 90 × 10 = 900 lb. = T_1. The effective tension $\mathbf{t} = T_1 - T = T_1\left(1 - \frac{T}{T_1}\right)$.

$H = \frac{t \times v}{33000}$, where H = the horse power and **v** = velocity of belt in feet per minute. [**v** = diameter of pulley in feet × π × revolutions per minute.]

In example 1, $\mathbf{t} = \frac{900 \times 14}{24}$; $\mathbf{v} = 600\,\pi$ for 60 revolutions of 10′ pulley, H = 30.
Example 2, $\mathbf{t} = \frac{0 \times}{3}$; $\mathbf{v} = 600\,\pi$ for 120 revolutions of 5′ pulley, $H = 34\frac{2}{7}$.
Example 3, $\mathbf{t} = \frac{900 \times 25}{35}$; $\mathbf{v} = 600\,\pi$ for 150 revolutions of 4′ pulley, $H = 36\frac{36}{49}$.

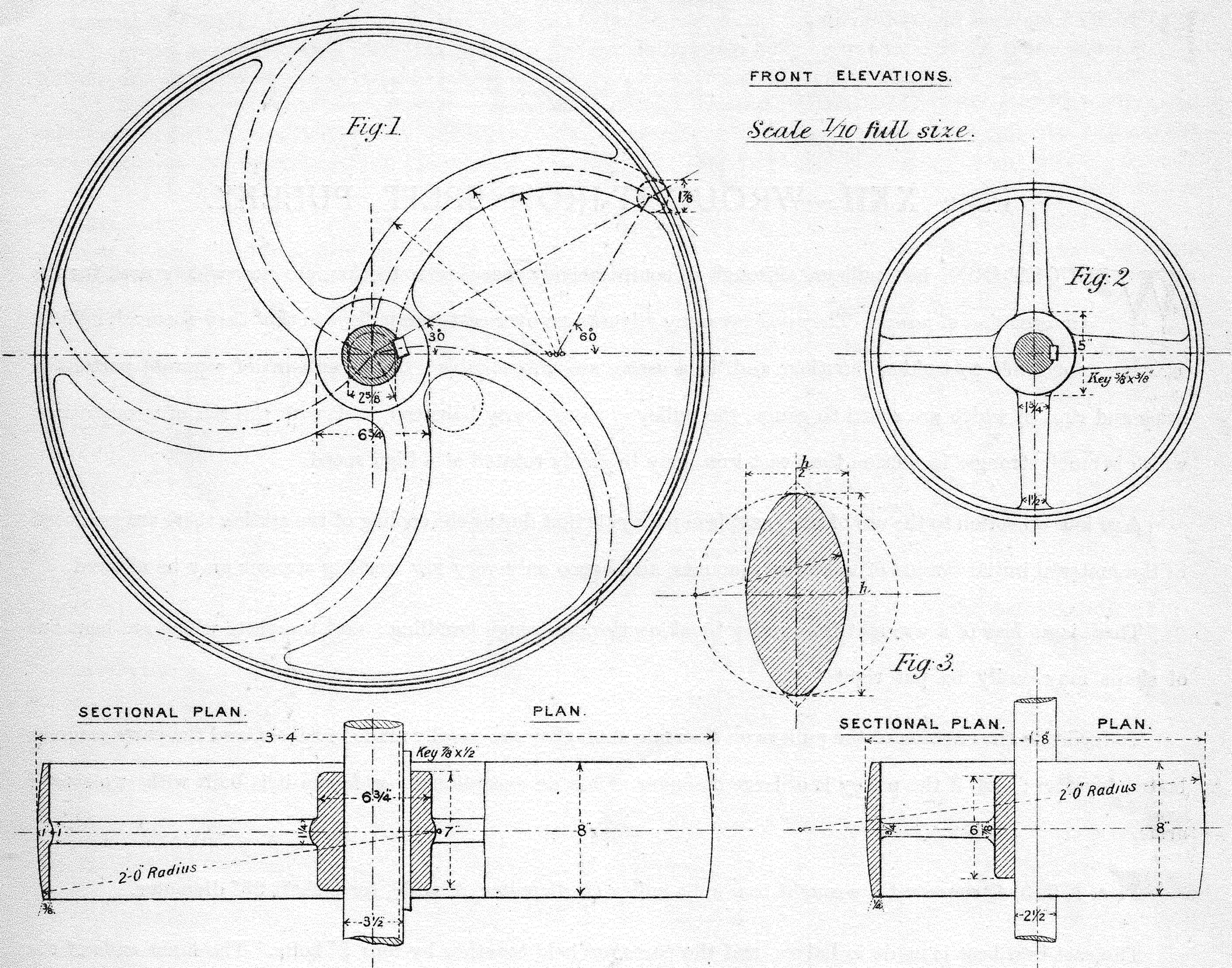
A rough approximate rule, which is sometimes used, is given below. It is based upon the assumption that a belt, running at 1000 feet per minute, will transmit 2 horse power for each inch of width :—

$$H = {\cdot}006\ D\ w\ R,$$

where D = diameter of pulley in feet; **w** = width of belt in inches; R = number of revolutions per minute.

This rule gives for the above examples, which are all for the same width and speed of belt, H = 36. This is approximately the power determined above for pulleys of the same diameter.

CAST IRON BELT PULLEYS.



CONE OR SPEED PULLEY.

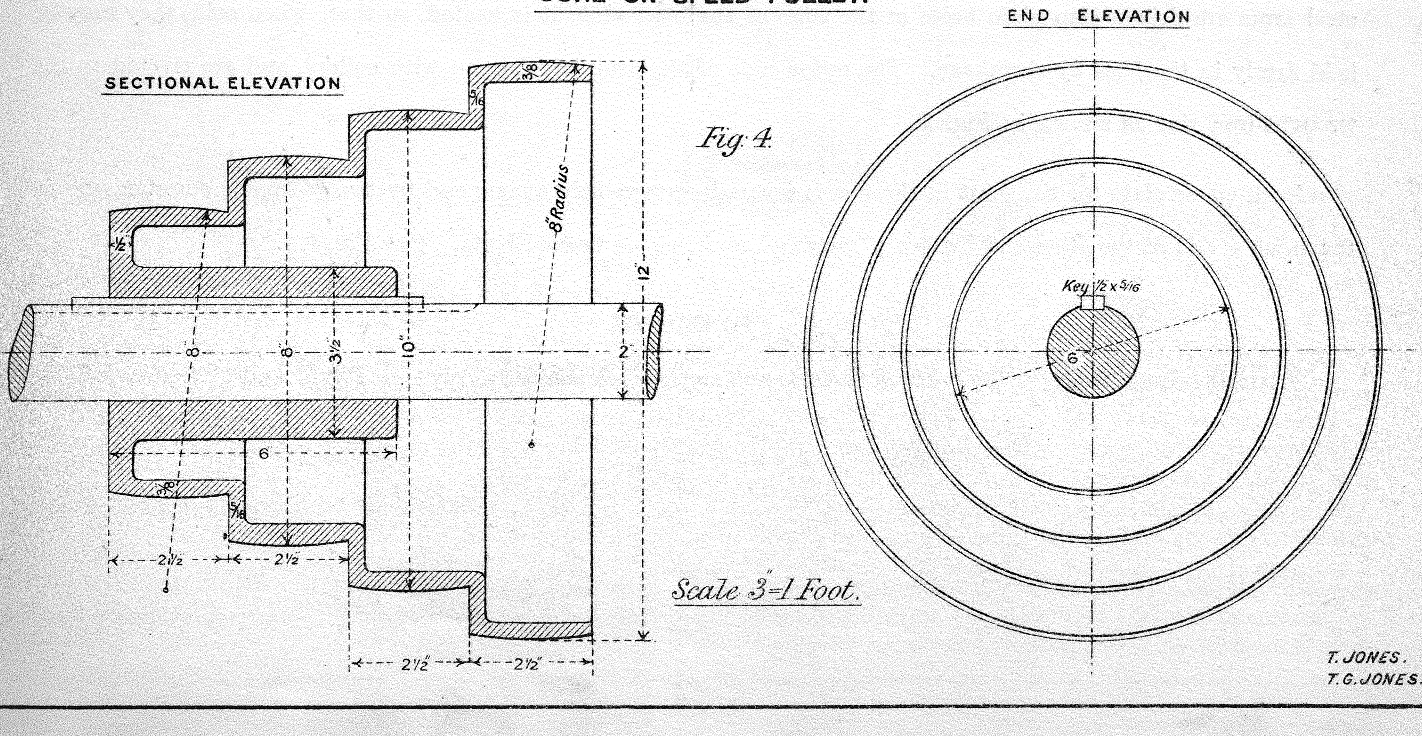


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Plate XXII.—WROUGHT-IRON SPLIT PULLEY.

WROUGHT-IRON belt pulleys, although of comparatively recent introduction, are now widely used for the transmission of power. They possess many advantages over cast-iron pulleys in that they are much lighter, stronger, not liable to sudden bursting, and, in a sense, are unbreakable Being built up of separate parts, the arms and rims of which are rolled to gauge, the pulley is more nearly balanced, and, with the rim of wrought-iron which is much stronger in tension than cast iron, may be safely rotated at a high speed.

A serious objection to the use of large cast-iron pulleys is that during the cooling of the casting there are produced in the material initial strains of unknown amounts, and hence only very low working stresses may be allowed.

There is no fear of a wrought-iron pulley breaking through rough handling ; and the parts which are bent out of shape may easily be put right.

To facilitate the erection of the pulleys on the main shaft they are usually made in halves, and the parts securely bolted together ; and, if the pulley is of large diameter or has an exceptionally wide rim it is built with two sets of arms.

Figs. 1, 2, 3, 4, represent a wrought-iron split pulley, 2′ diameter, 5″ wide, for a shaft $2\frac{3}{4}$″ diameter.

The cast-iron boss is made in halves, and the parts are held together by four $\frac{5}{8}$″ bolts. The inner ends of the steel arms are driven into plain holes in the edge of the boss while it is heated, so that, when cold, they may be held firmly in the boss by shrinkage. The outer ends of the arms are formed with collars, and are riveted to the wrought-iron rim as shown in Fig. 3.

Each cover plate for the joint in the rim is secured permanently at one end by two $\frac{3}{8}$″ rivets, countersunk on the outside, and at the other end by two $\frac{1}{2}$″ nuts and countersunk headed bolts. (See Fig. 4.)

EXERCISE.

Wrought-Iron Split Pulley.—Draw the side and sectional elevations as given in Figs. 1 and 2. *Scale $\frac{1}{2}$ full size.*

WROUGHT IRON SPLIT PULLEY.

Scale 3" = 1 Foot.

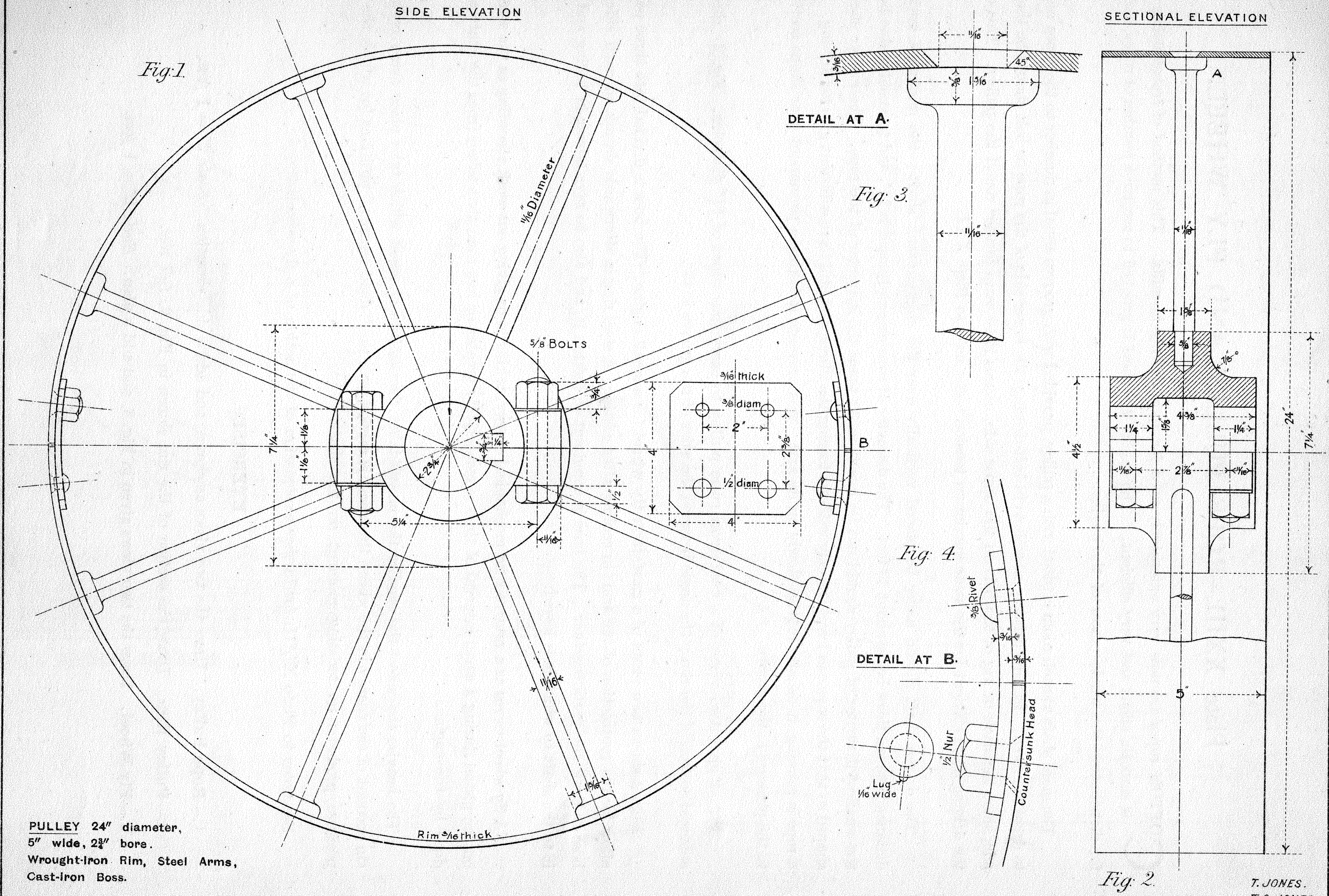

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Plate XXIII.—ROPE PULLEY AND FLY WHEEL.

COTTON ropes are now very extensively used for main driving in mills. This method of transmitting power is smooth, silent and very efficient. The engine fly wheel is grooved to receive a number of ropes, sufficient for distributing the whole power of the engine.

The main shaft to each room is fitted with a pulley, provided with grooves equal in number to the ropes required to drive the room. In rope driving, as well as in belt driving, the lower side of the rope or belt should be the tight side, since the slack or sag of the belt on the upper side, causes it to wrap further round the pulley. This makes the driving more efficient for the same maximum tension.—*See notes to Plate XXI.*

The driven shaft should never be placed vertically over the engine crank shaft; the minimum horizontal distance between them being about 20 feet. The diameter of rope generally used is $1\frac{3}{4}''$, and to avoid undue bending the smallest diameter of pulley should not be less than 30 times the diameter of the rope. The groove should be shaped so that the rope can leave it easily without tearing the fibres, and still have sufficient grip for driving. When a rope is running, it usually rotates slowly on its axis; this prevents it taking a permanent pear shape section due to the pressure in the groove.

Fig. 1 gives two views of a **cast-iron rope pulley** to carry four ropes of $1\frac{1}{2}''$ diameter. Fig. 2 shows an enlarged section of the rim; the angle between the sides of the groove is 45°.

Fig. 2 shows two views of a **cast-iron fly wheel** for a small engine. The boss is divided into three parts, to allow for the contraction of the arms in cooling after casting. A steel ring is afterwards shrunk on each side of the boss to draw the parts together. The boss is cored out to lighten it, and also to save labour in boring and fitting it to the shaft to which it is keyed.

A fly wheel serves as a reservoir for mechanical energy, taking in or giving out work according as the work done on the piston during a small interval of time is respectively greater or less than that taken from the shaft as work done by the engine.

The heavier the wheel the smaller the unavoidable fluctuations of speed which take place every revolution. On account of the energy in the wheel, the crank is carried over the "dead centres," positions for which the piston pressure produces no twisting moment on the shaft.

See notes to Plate XLIII. for further explanation of the action of a fly wheel.

EXERCISES.

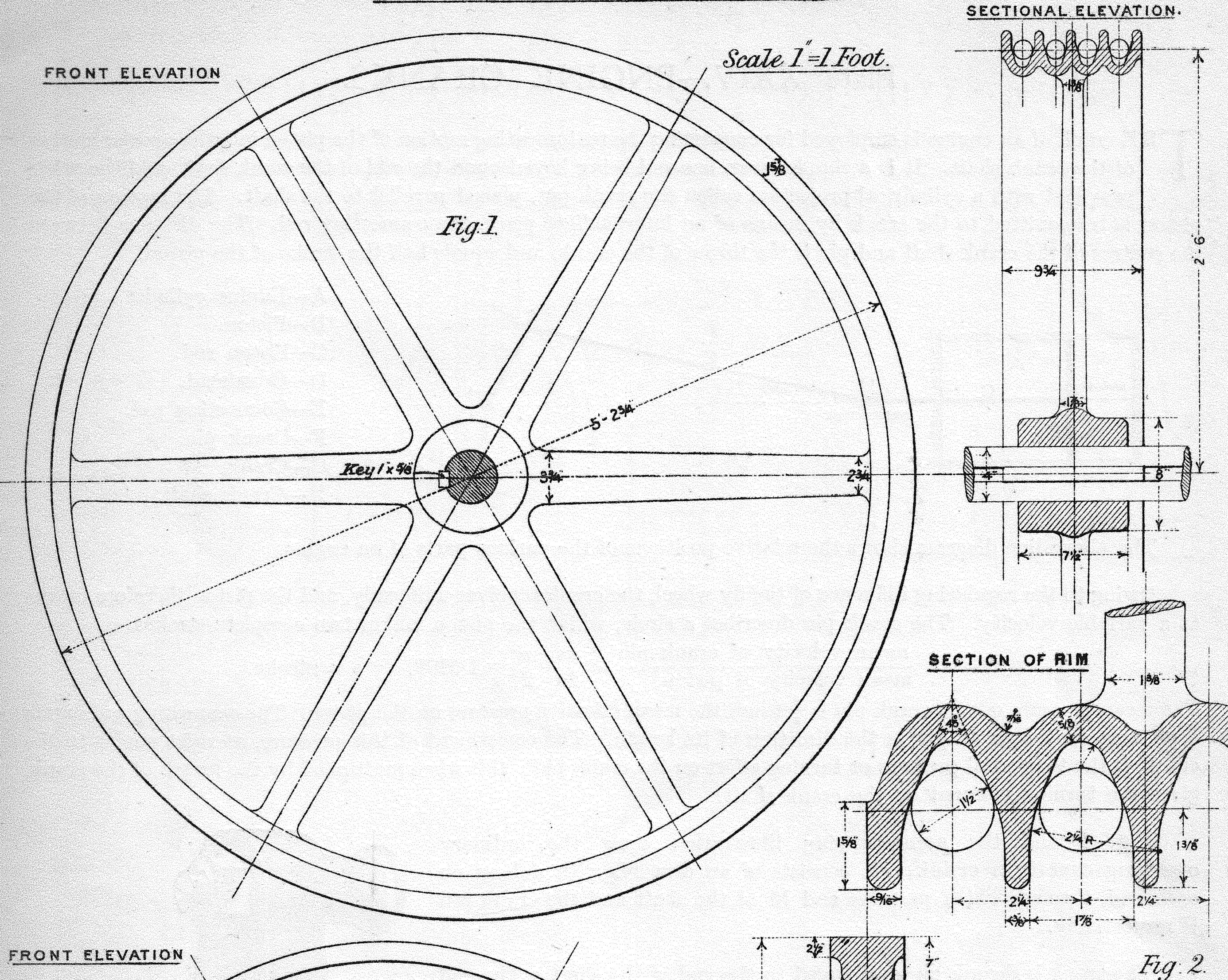
1.—**Rope Pulley.** Draw the two views as shown, and add a side elevation. *Scale* 2″ = 1 *foot.*

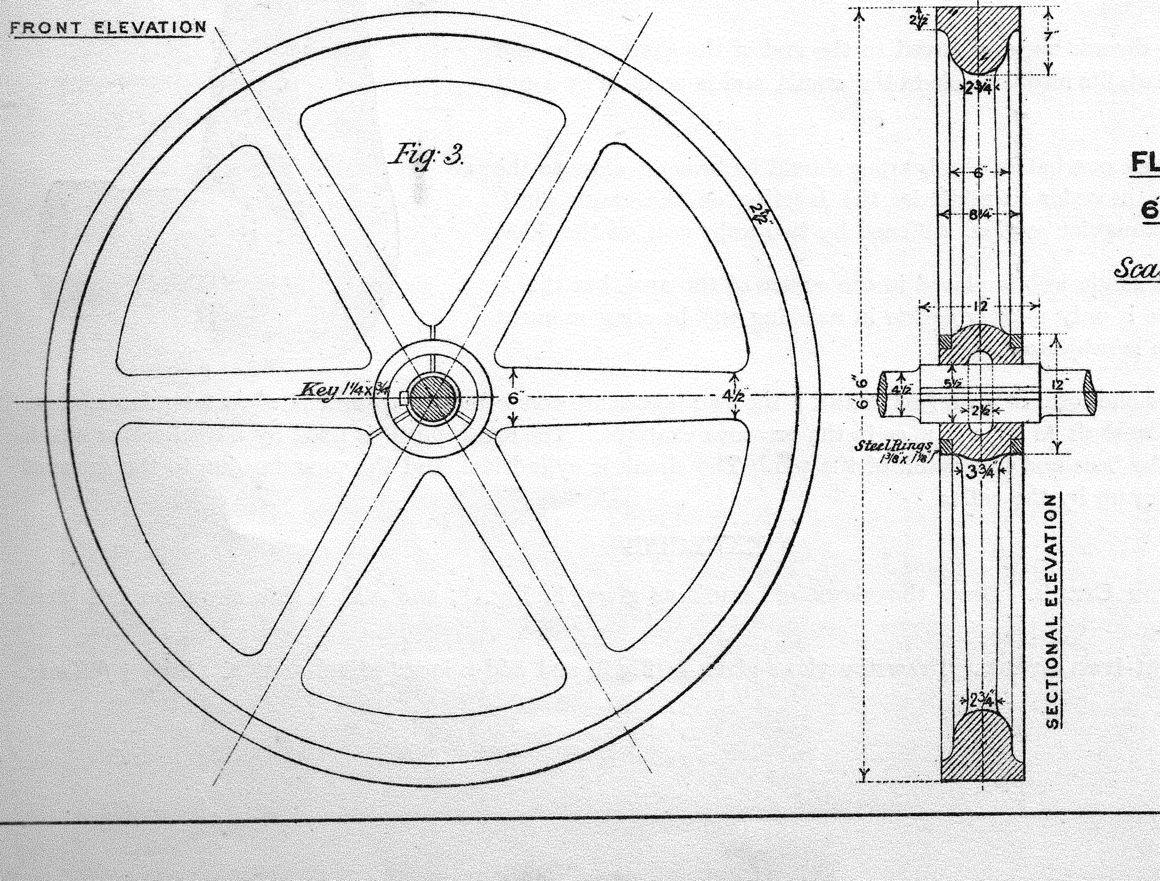
2.—**Pulley Rim.** Draw the section of the rim shown in Fig. 2, *full size.*

3.—**Fly Wheel.** Draw the two views given in Fig. 3, and add a plan. *Scale* 2″ = 1 *foot.*

ROPE PULLEY 5 FEET DIAMETER.

Scale 1" = 1 Foot.





FLY WHEEL.

6'-6" DIAM^R.

Scale ⅝" = 1 Foot.

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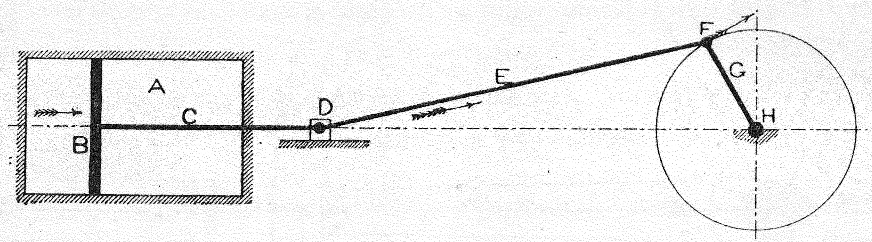
Plate XXIV.—ENGINE CRANKS.

THE crank of an engine is employed for converting the reciprocating motion of the piston into the circular motion of the crank shaft. It is a simple lever, one end being keyed upon the end of the crank shaft, and the other provided with a cylindrical projection called the crank pin, placed parallel to the shaft. The motion of the piston is transmitted to the crank by means of an intermediate piece, the connecting rod. The distance between the centres of the crank shaft and pin is the throw of the crank, and equals half the stroke of the piston.



A—Engine cylinder.
B—Piston.
C—Piston rod.
D—Crosshead.
E—Connecting rod.
F—Crank pin.
G—Crank.
H—Crank shaft.

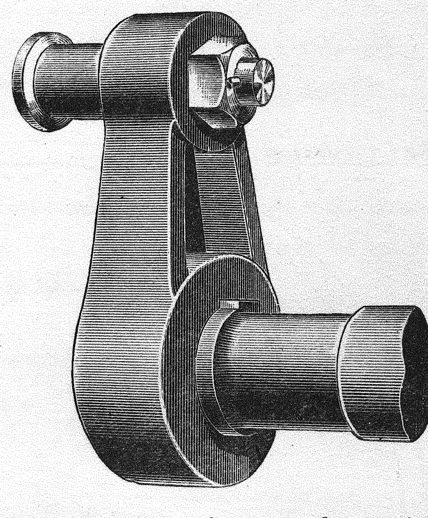
The above line diagram shows the relative positions of the various parts of an engine.

Owing to the regulating influence of the fly wheel, the crank revolves uniformly, and the piston therefore moves at a variable velocity. The crank pin describes a circle, whilst the piston makes two complete strokes.

$$\therefore \frac{\text{mean velocity of crank pin}}{\text{mean velocity of piston}} = \frac{\pi s}{2s} = \frac{\pi}{2} = 1{\cdot}5708. \qquad s = \text{stroke}.$$

The mean pressure on the crank pin is $\frac{2}{\pi}$ times the mean effective pressure on the piston. The connecting rod exerts a pressure on the crank pin in the direction of its length. The component of this pressure, at right angles to the crank, is the tangential pressure or turning effort on the crank pin; this when multiplied by the length of the crank, gives the turning moment on the crank shaft.

Fig. 1 and the accompanying illustration show the ordinary overhung **cast-iron crank.** It consists of an arm N, with a boss on each end, through which pass the end M of the shaft and the crank pin P respectively.



The crank is shrunk on, and keyed to the end of the shaft. The end of the pin is conical, fits into the hole in the crank, and is secured by a nut and taper pin.

Since the crank overhangs, the bearing should be brought close to the face of the crank, in order to minimise the bending on the crank shaft. The arm is of rectangular section, stiffened by two webs cast on the sides.

Sometimes a single web is placed in the centre of the arm, but this is not so good, since it only slightly assists in resisting the bending moment to which the arm is subjected.

Fig. 2 shows one form of a **forged crank**; the two bosses are connected by an arm N of rectangular section. It is secured to the shaft M by a key, as in the previous example. The crank pin P is fixed by a steel cotter which passes through the boss and the shank of the pin. The shank is cylindrical, and the pressure due to the draw of the cotter is taken up by the collar.

EXERCISES.

1.—**Cast-iron Crank.** Draw the views of crank as given in Fig. 1, and add a side elevation and plan. *Scale $\frac{3}{8}$ full size.*

2.—**Wrought-iron Crank.** Draw the views given in Fig 2, and add a longitudinal section. *Scale $\frac{1}{4}$ full size.*

ENGINE CRANKS.

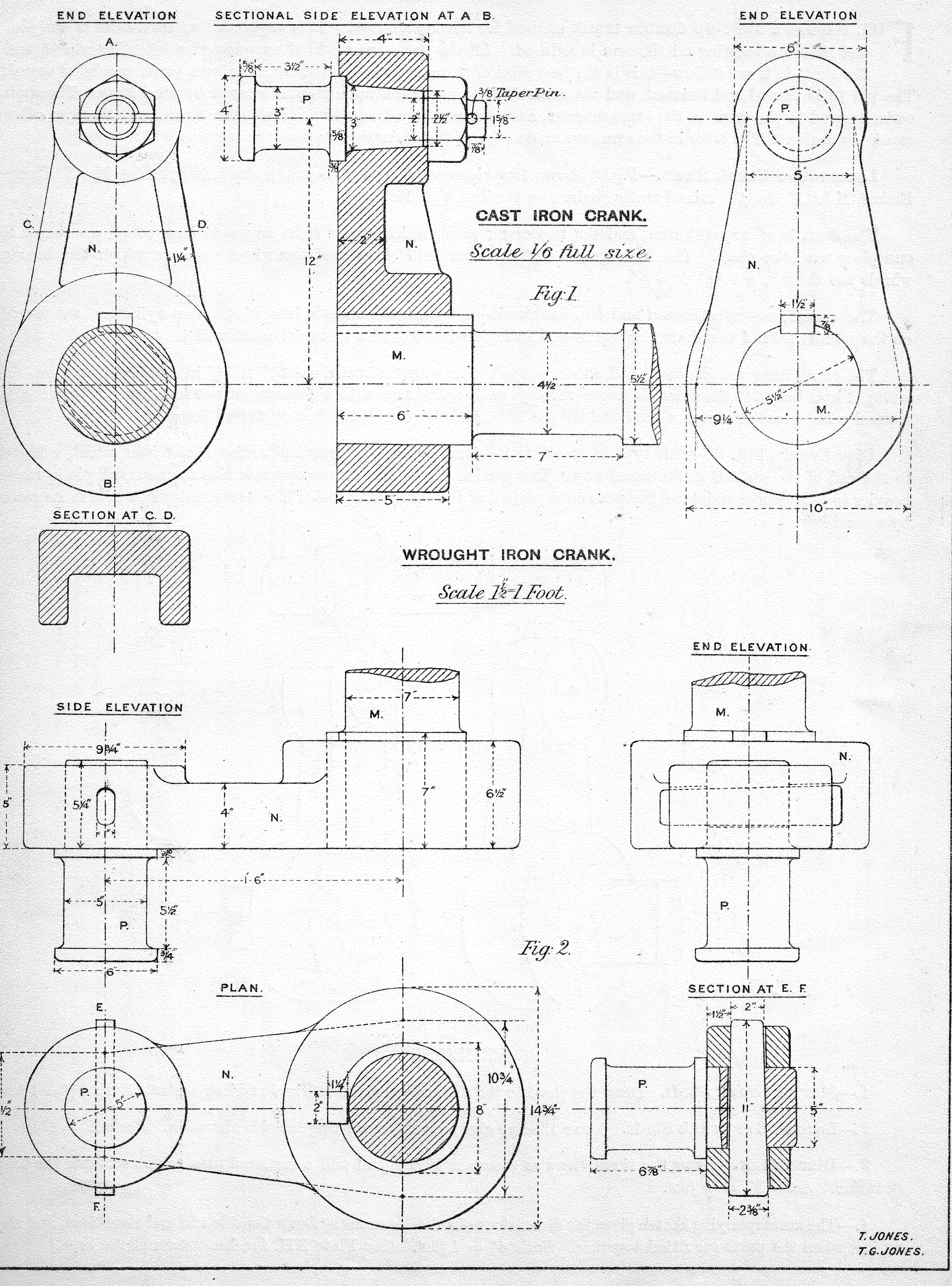


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Plate XXV.—ENGINE CRANKS AND CRANK SHAFTS.

FIG. 1 shows a **built-up double crank** as used for marine engines. It is supported on both sides of the pin, and thus the twisting on the arm is reduced. All the parts are made of compressed steel. The pin M and the shaft L L are hollow—this is the best section to resist twisting and bending for a given weight of metal. The pin M is turned and finished, and the arms N N then shrunk upon it, and secured by steel keys. The shaft ends, secured to the arms in the same manner, have flanges forged on them, which serve to connect them to other cranks or shafts. The holes in the arms are made $\frac{1}{1000}$ of the diameter less than the parts to which they fit.

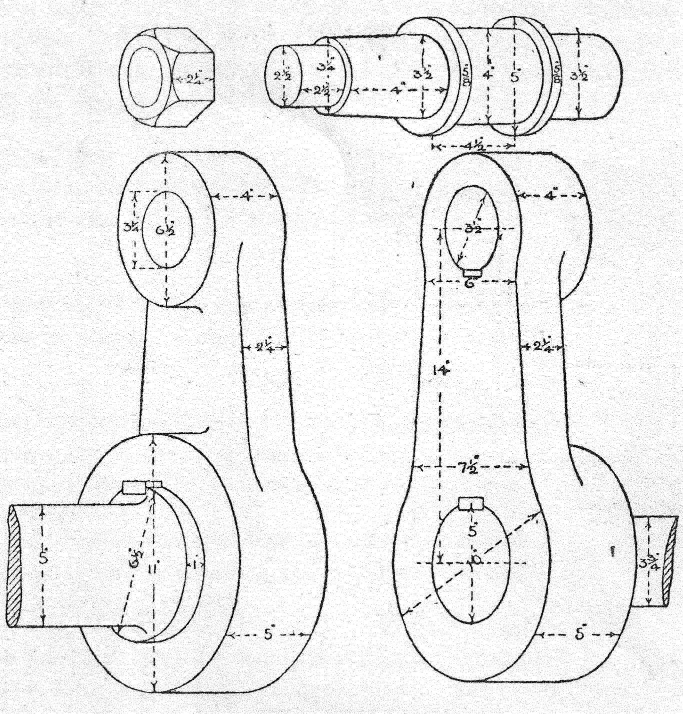
Locomotive Crank Shaft.—Fig. 2 shows two views of a locomotive crank shaft designed by Mr. T. Hurry Riches, M.I.C.E., for the mixed traffic engines on the Taff Vale Railway.

The shaft is of wrought iron, and the two crank arms, which are at right angles to each other, are forged in one piece with the shaft. The bearings are $7\frac{1}{2}''$ diameter $\times$ $7''$ long, and the wheel seats, on which the driving wheels are shrunk, are $8\frac{5}{8}'' \times 7\frac{5}{16}''$.

The eccentrics—two forward and two backward—actuating the slide valves of the two cylinders, are placed on the middle part of the shaft. (See Plate XXVI., for drawing of a locomotive eccentric).

The crank arms are elliptical, and all are hooped with wrought-iron rings, $2\frac{1}{2}'' \times 1\frac{1}{2}''$ in section, shrunk on. A wrought-iron bolt, $2\frac{1}{4}''$ diameter, is forced through each crank arm with a pressure of not less than 25 tons ; it is secured with a cast-steel nut with 8 threads per inch, and the end of the bolt is riveted over.

Disc Crank, Fig. 3.—This type of crank, as its name implies, is formed of a disc of cast iron, which is keyed to the end of the shaft S in the usual way. The portion of the disc on the opposite side to the crank pin is made heavier to balance the weight of the pin and a portion of the connecting rod. The steel crank pin is held in its place by a steel key.



EXERCISES.

1.—**Marine Crank Shaft.** Draw the shaft to the dimensions given in Fig. 1, adding a plan. *Scale* $1\frac{1}{2}'' = 1$ *foot.*

2.—**Locomotive Crank Shaft.** Draw the two given views, completing the side elevation. *Scale* $2'' = 1$ *foot.*

3.—**Disc Crank.** Draw the three views as shown in Fig. 3, and add a sectional plan projected from the back elevation. *Scale* $3'' = 1$ *foot.*

4.—The accompanying sketch gives the dimensions of a double crank. Draw the side and end elevations, and also a plan, when the parts are fitted together. *Scale* $4'' = 1$ *foot.* (See Plate XII. for dimensions of the keys.)

ENGINE CRANKS AND CRANK SHAFTS.

Scale 1/20 full size.

MARINE CRANK SHAFT.

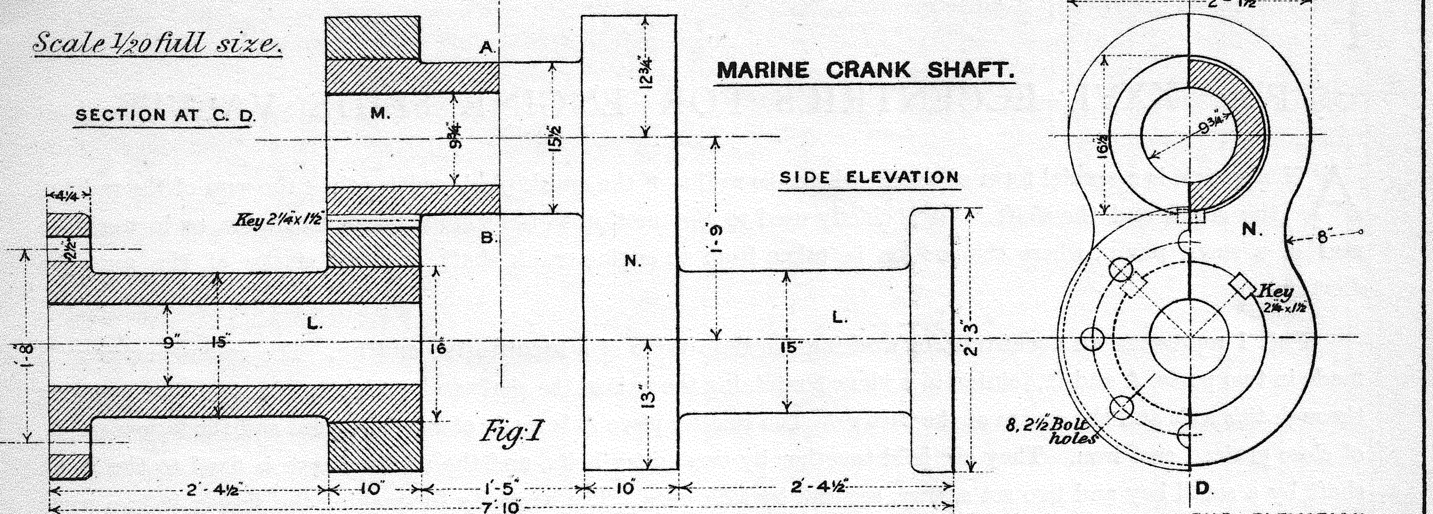


Fig: 1

LOCOMOTIVE CRANK SHAFT

Scale 1" = 1 Foot.

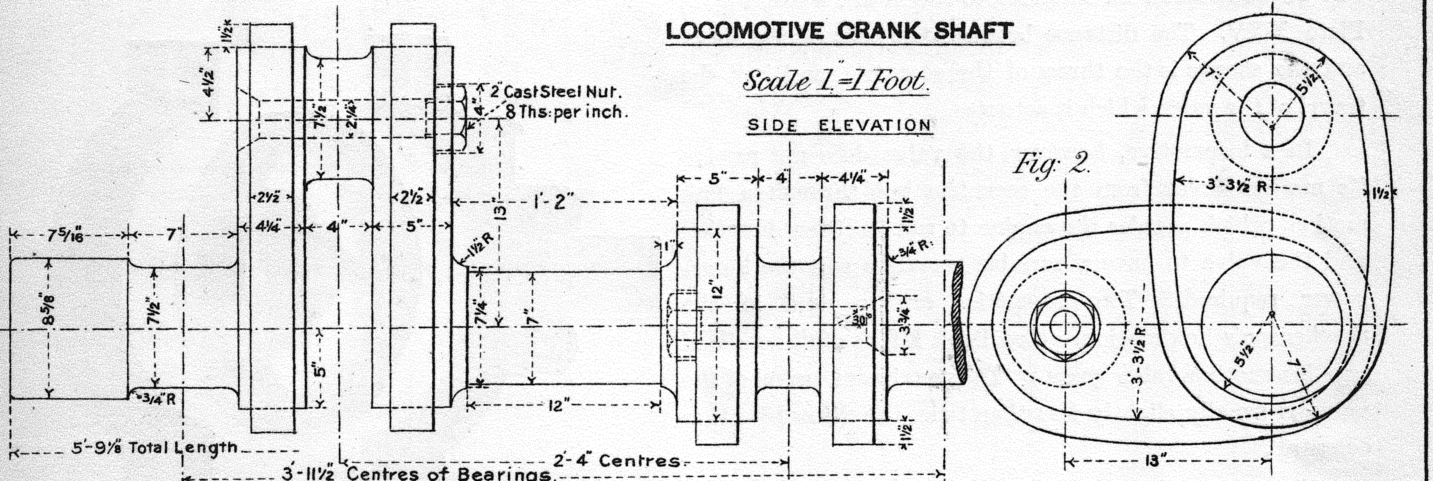


Fig: 2.

CAST IRON DISC CRANK.

Scale 1/10 full size.

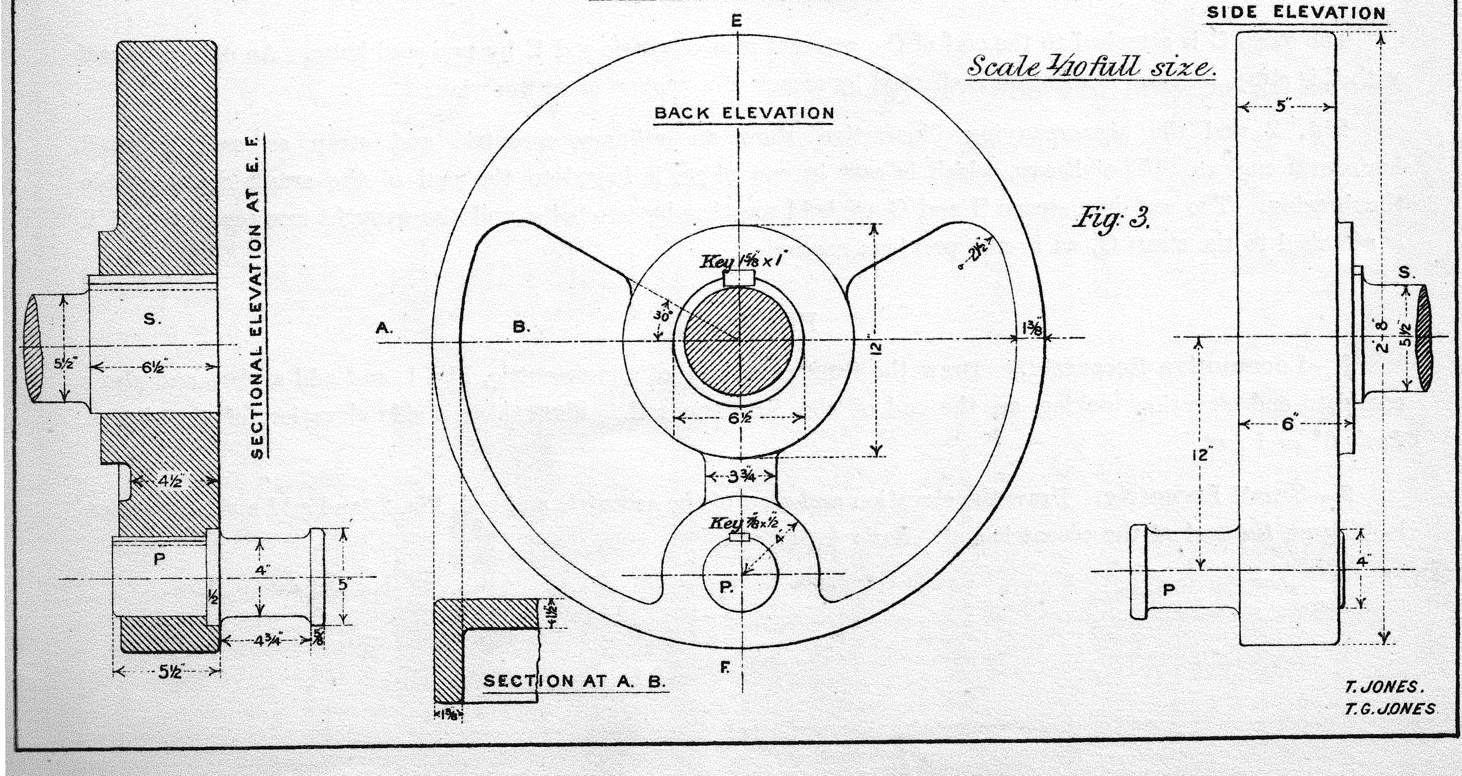


Fig: 3.

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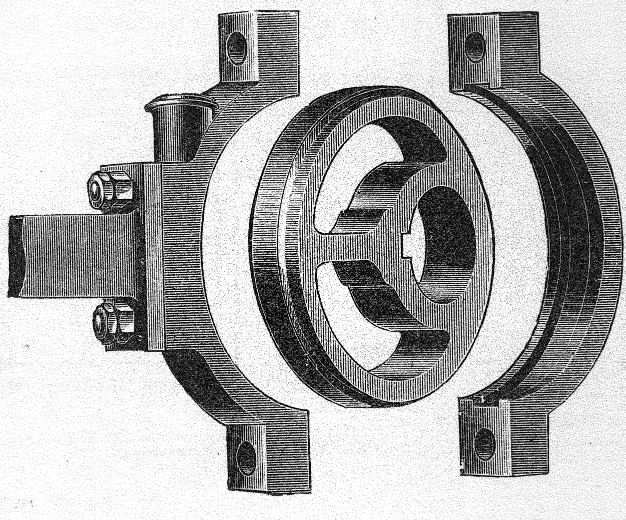
Plate XXVI.—ECCENTRICS FOR ENGINE SLIDE VALVES.

AN eccentric is a special form of crank, where the radius of the crank pin is greater than the sum of the radii of the crank and the shaft. It is chiefly used to give motion to the slide valve of an engine, or to work the ram of a small pump, where the motion is taken from an engine crank shaft, and the stroke of the pump a short one.

Fig. 1 represents an **eccentric** for working the slide valve of a **locomotive engine.** The eccentric sheave is made in two parts, A and B, jointed in a plane containing the axis of the shaft and at right angles to a plane passing through this axis and the centre of the sheave. The smaller piece A is made of wrought iron, and the larger one B, of close grained cast iron. They are held together by two cotter bolts, and the whole sheave is fixed to the crank shaft by a small key and two set screws, secured by lock nuts. The necessity for making the sheave in two parts will be understood on referring to the crank shaft, Fig. 2, Plate XXV. The distance between the axes of the shaft and the sheave is the throw of the eccentric, and half the travel of the valve which it actuates.



In a locomotive, however, the valve does not receive its motion direct from the eccentric but through a link motion, which enables the engine to run in either direction and the valve to have a variable travel to suit the horse-power required. Two eccentrics are required in the Stephenson's link motion. (See Book III. for full details of two locomotive valve gears.) The maximum travel of the valve is usually slightly less than twice the throw of either eccentric.

The straps C and D, which embrace the sheave, are made of cast iron and are held together by the bolts H H.

The strap C is attached to the end of the wrought-iron eccentric rod E by two stud bolts. An oil box is cast with this strap, and the lubrication is effected by means of a syphon of cotton wick.

Fig. 2 and the accompanying illustration show an ordinary eccentric and strap as used in small horizontal engines. The sheave, which is cast in one piece, is keyed to the end of the crank shaft outside the bearing. The cast-iron straps B and C are held together by two bolts, and the wrought-iron eccentric rod E is attached to the strap C, as in the previous example.

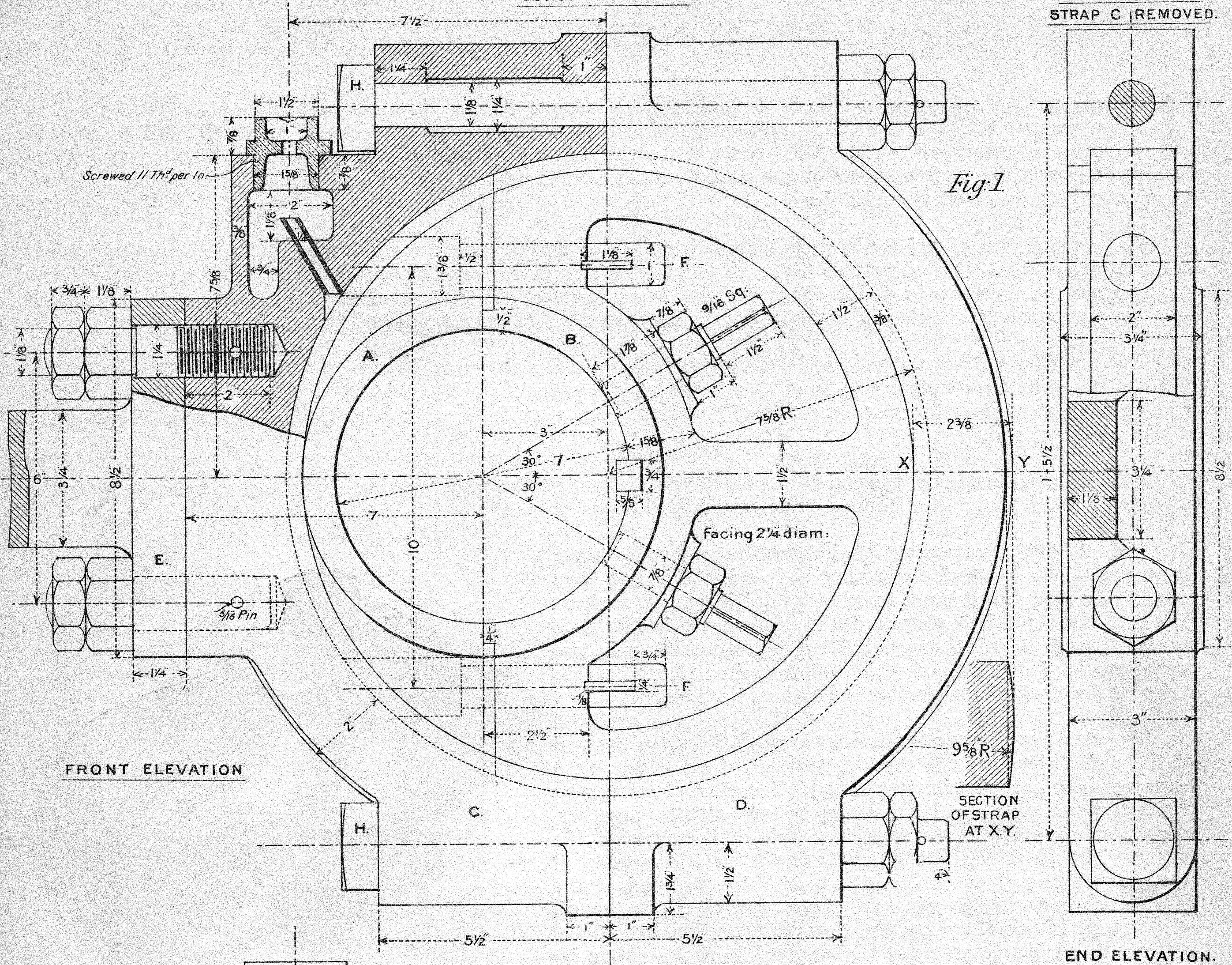
EXERCISES.

1.—**Locomotive Eccentric.** Draw the front elevation of the eccentric, Fig. 1, and add a plan, and also a complete end elevation looking on the end of the eccentric rod. Show also a side elevation of the sheave. *Scale 6″ = 1 foot.*

2.—**Small Eccentric.** Draw the elevation and plan of the eccentric and rod, Fig. 2, and add a side elevation looking on the end of the eccentric rod. *Scale ½ full size.*

ECCENTRIC FOR LOCOMOTIVE ENGINE.

Scale 3" = 1 Foot.



ECCENTRIC AND ROD FOR HORIZONTAL ENGINE.

Scale 1/5 full size.

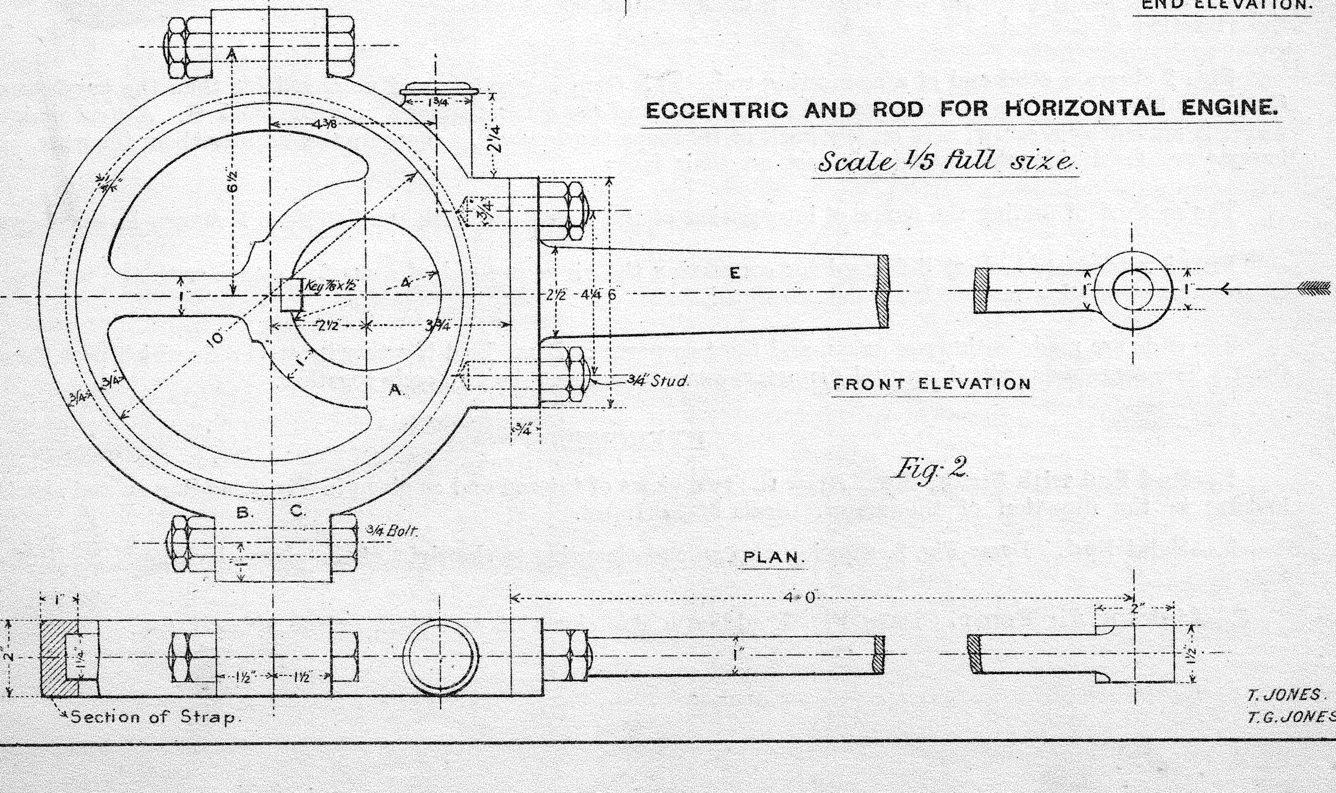


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Plate XXVII.—CONNECTING ROD ENDS.

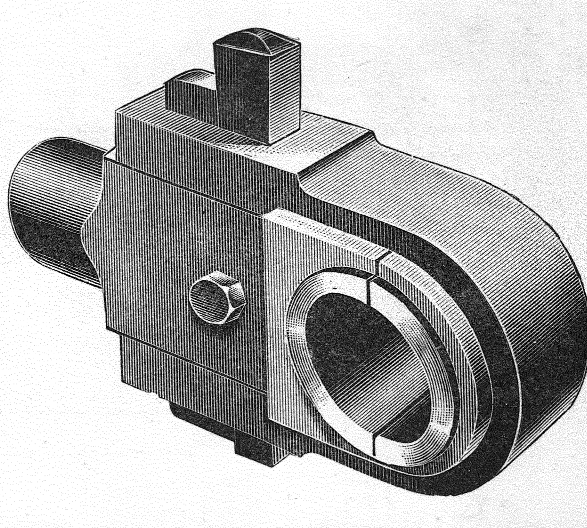
THE connecting rod of an engine is the link which connects the crosshead to the crank pin. By its use, in conjunction with the cross head and crank, the rectilinear motion of the piston is converted into the circular motion of the crank shaft. The length of the rod, from the centre of the crosshead pin to the centre of the crank pin, should, if possible, never be less than four times the length of the crank arm, since a short rod produces an excessive pressure on the slide bars.

The usual length of rod for land engines, is from five to seven times the crank arm. The two ends of the rod are accurately fitted with adjustable bearings, to take up the wear. The crosshead pin is smaller than the crank pin, because the former is in double shear, and the latter is sometimes supported at one end only, and is subjected to a bending moment. This is the reason for the difference in the construction of the two ends of the rod.

A connecting rod has not only to be stiff enough to resist the alternate thrust and pull to which it is subjected, but also to resist the tendency to bend due to its rapid oscillations. The section of the rod is usually circular, having its greatest diameter near the centre of its length, or else, gradually increasing in diameter from the crosshead to the crank pin end.

In a locomotive engine, the rod is rectangular in section, the longer side being in a plane at right angles to the axis of the crank pin. (See Plates XXXI. and XXXII.)

Fig. 1, and the perspective illustration, show one form of wrought-iron or steel connecting rod end. This has been extensively used, but it is not adapted for quick running engines. The end of the rod E is rectangular in section, and is slotted to receive the gib B and the cotter A. The brasses D are in two parts, one having a flat end which butts against the end of the rod, and the other, a semicircular end fitting into the strap C.



The strap passes round the brasses, and fits upon the end of the rod. Slots are cut through the two sides of the strap corresponding with that in the rod end. The gib and cotter pass through these slots, and draw the brasses tightly together, clearance being left in the slots to admit of the *draw* of the cotter. The heads on the gib fit exactly on the outside of the strap, and so keep it in contact with the flat end of the rod. The wear, which is principally in the direction of the length of the rod, is taken up by the downward movement of the cotter. A set screw prevents the slackening of the cotter by the vibration of the rod.

Fig. 2 shows a solid end of a connecting rod. This form is stronger and more reliable than the previous one. One brass fits against the semicircular end of the hole, and the other against a movable block B. A steel wedge A, which is moved by a screw, acts on the back of B, keeps the brasses in contact, and adjusts them for wear. The brasses are fixed sideways by flanges cast on their faces.

The method of finding the curve of intersection of the round end with the flat sides is shown in the diagram.

Fig. 3 represents a short link used for connecting the air pump rod of a marine engine with the lever which receives an oscillating motion from the crosshead.

The ends are made entirely of brass, and the two parts D D are held together by steel rods which pass through them. The same rods, with increased diameter, serve to connect the two ends together.

EXERCISES.

1.—**Rod End with Strap, &c.** Draw the two views of the rod end as given in Fig. 1, adding an end elevation looking in the direction of the arrow. *Scale* 9″ = 1 *foot*.

2.—**Solid End.** Draw Fig. 2, showing also an end elevation to the right of the given elevation. *Scale* 6″ = 1 *foot*.

3.—**Link for Air Pump.** Draw Fig. 3, adding a side elevation and plan. *Scale* 9″ = 1 *foot*.

CONNECTING ROD ENDS.

Scale 1/3 full size.

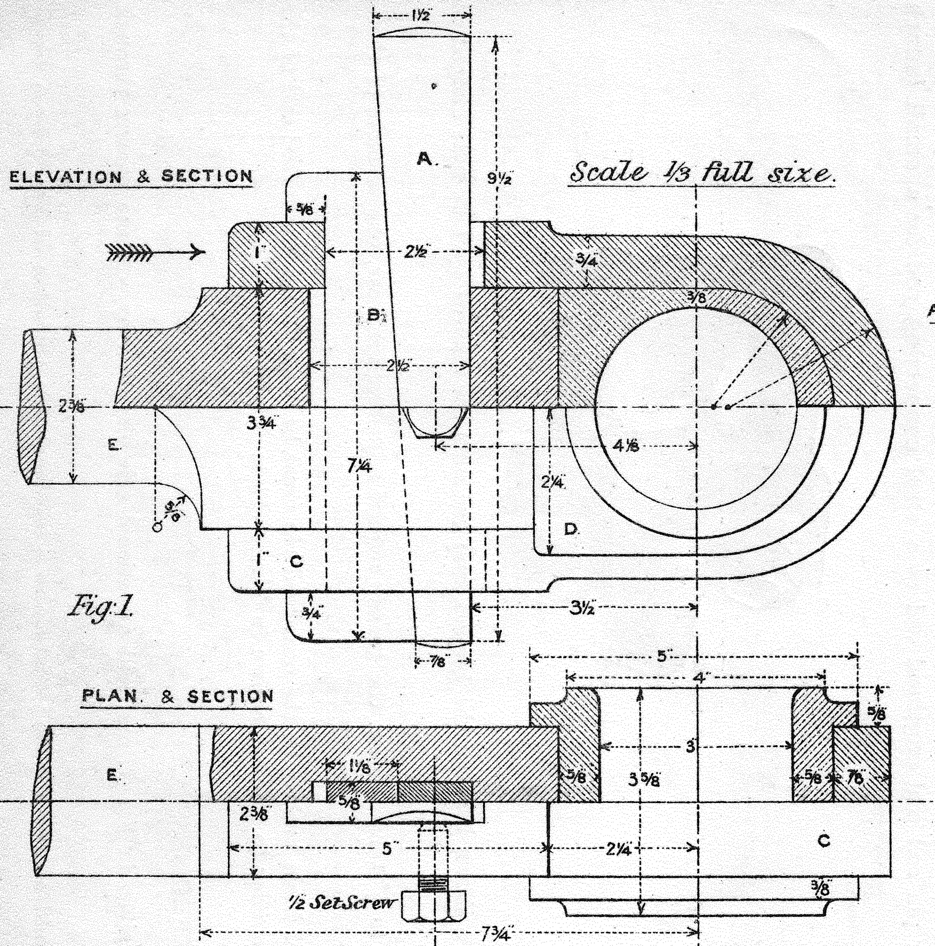


Fig. 1.

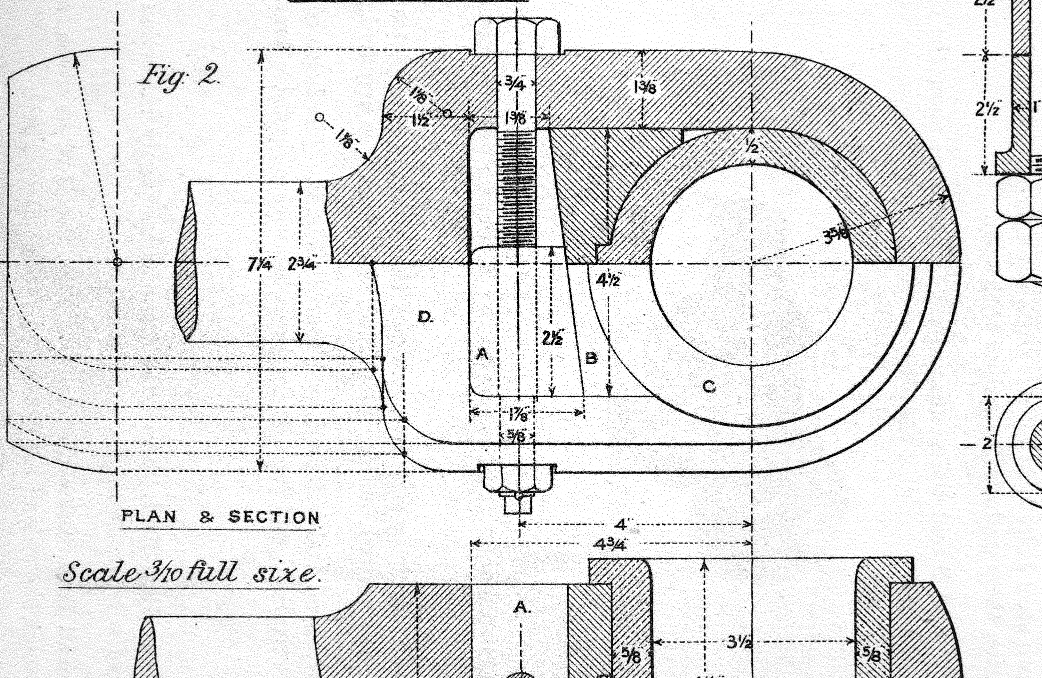


Fig. 2.

Scale 3/10 full size.

LINK FOR AIR PUMP.

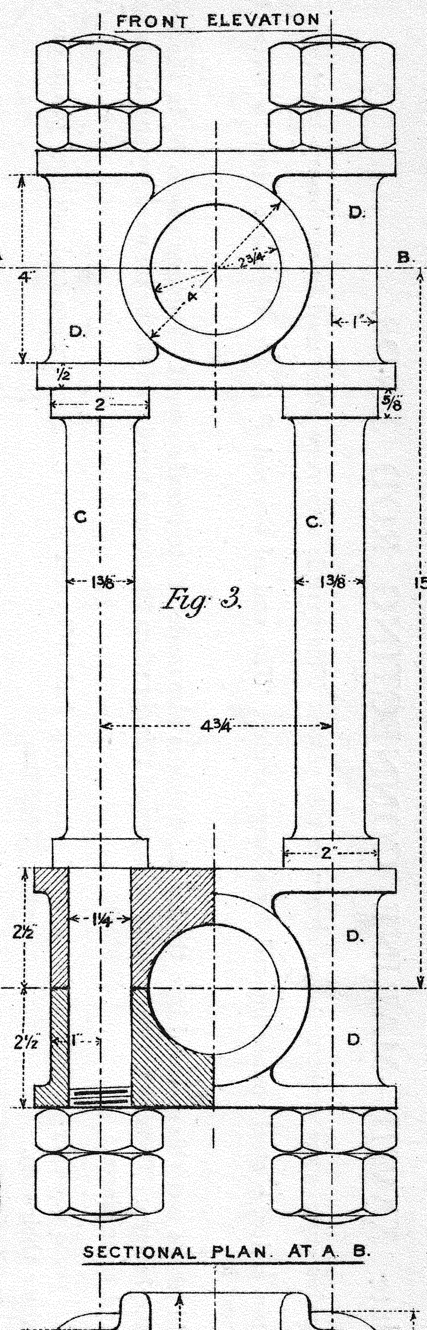


Fig. 3.

Scale 3" = 1 Foot.

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Plate XXVIII.—MARINE CONNECTING ROD ENDS.

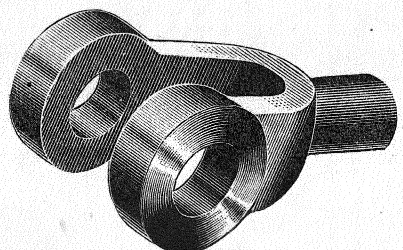
CONNECTING rod ends for marine engines are constructed as shown in this plate. This type is also very often used in stationary engines.

Fig. 1 shows the crank pin end of a steel connecting rod for a large engine. The end of the rod P is forged in one piece, a recess being planed across it to receive the brass M.

The bottom and top brasses M M fit into this end and into the cap L, which is also made of steel and recessed. Packing pieces N, in the form of the letter U, are placed between the brasses. These can be removed, and planed thinner when the brasses require adjusting for wear, by simply slackening the bolts without taking them out of their places.

The two steel bolts hold all the parts together as shown, the grooved nuts being locked by set screws. The brasses and packing pieces are cored out to lighten them and also to reduce their cost.

Figs. 2 and 3 and the accompanying illustrations show the crank pin end and crosshead end of a **steel connecting rod** for a **small launch engine.**



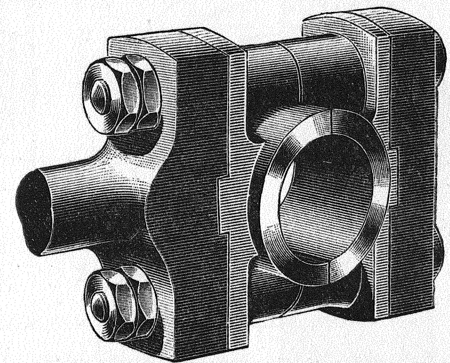


Fig. 2 is similar in construction to Fig. 1, but no packing pieces are shown between the brasses, and the nuts are secured by lock nuts instead of set screws.

The packing pieces are often made of thin wrought-iron plates $\frac{1}{32}''$ to $\frac{1}{8}''$ thick: when the brasses require adjusting for wear, one or more of these plates can be removed, and no planing of the packing is required.

The crosshead end is forked, as shown in Fig. 3. The holes are not brass bushed, so the crosshead pin must be made fast in the forked end of the rod. The crosshead will have to be provided with adjustable brasses similar to the one shown in Fig. 1, Plate XXIX.

EXERCISES.

1.—**Large Rod End.** Draw the three views partly in section as given in Fig 1, and add a sectional side elevation by a plane passing through the centre of a bolt. *Scale* 6″ = 1 *foot.*

2.—**Small Rod.** Draw the views of Figs. 2 and 3, as given, and add a plan of each; show also a sectional side elevation of Fig. 2 by a plane passing through the axis of the crank pin. *Scale* 6″ = 1 *foot.*

CONNECTING ROD ENDS FOR MARINE ENGINES.

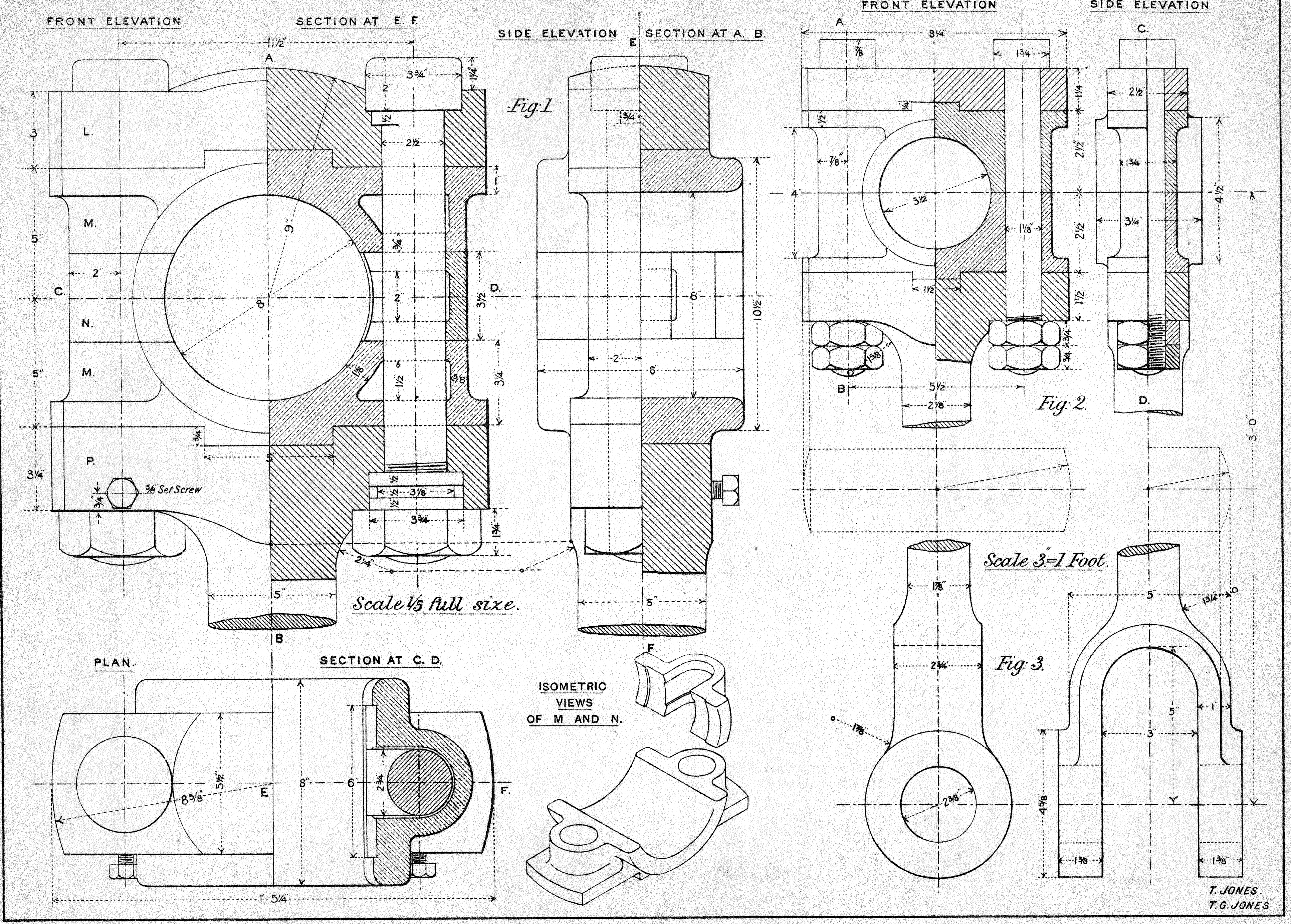


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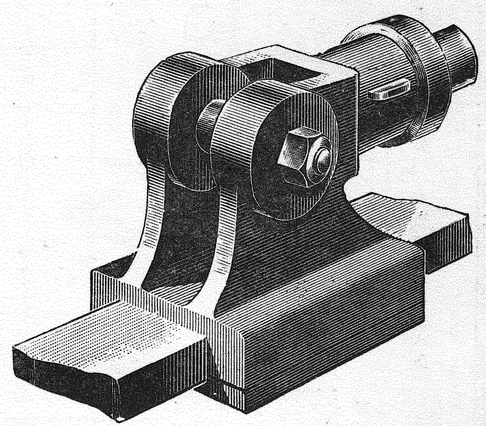
Plate XXIX.—ENGINE CROSSHEADS.

ENGINE Crossheads, in conjunction with slide bars, are used for guiding the piston rod in a straight line, and for resisting the thrust or pull of the connecting rod, when it is inclined to the line of direction of the piston rod. (See line diagram of engine shown in notes to Plate XXIV.) The end of the piston rod is usually turned to fit a hole bored in the crosshead, to which it is secured by a cotter.

The end of the connecting rod is also fitted to the crosshead: a round pin passing through both connects them together and admits of the oscillating motion of the connecting rod. The slide bars are either cast as part of the engine bed or are bolted firmly to it.

Crosshead for Vertical Engine.—Fig. 1 shows a steel crosshead for a vertical engine. The piston rod A has a T end to which the slide block B containing the brasses C is secured by the two steel bolts and cap D. This arrangement admits of adjustment for wear of the brasses. The base of the block B moves in slides which are bolted fast to the engine bed.

Crosshead for Horizontal Engine.—Fig. 2 and the accompanying illustration show a form of cast-iron crosshead used for small horizontal engines.



The end of the piston rod C is made slightly conical, and is held firmly in the hole of the crosshead by a steel cotter.

The body of the crosshead A is made of box section. The end of the connecting rod fits into the recess in the crosshead, and the pin D connects them together.

The base of the crosshead is fitted to the slide bar B, upon which it can move freely. The drawing shows the crosshead placed in position on the slide bar. The bottom cast-iron plate, secured by six small set screws, keeps the crosshead down on the bar. There is no stress on this plate when the engine is working, since the pressure, due to the obliquity of the connecting rod, is on the top side of the bar.

The pin D is made conical where it fits into the sides of the crosshead, but the part to which the connecting rod end is attached is cylindrical.

EXERCISES.

1.—**Crosshead for Vertical Engine.** Draw the three given views, completing the plan. *Scale* 9″ = 1 *foot*.

2.—**Crosshead for Horizontal Engine.** Draw the three given views and add an end elevation looking on the end of the piston rod. *Scale* 6″ = 1 *foot*.

ENGINE CROSSHEADS.

CROSSHEAD FOR VERTICAL ENGINE.

Scale 3/10 full size.

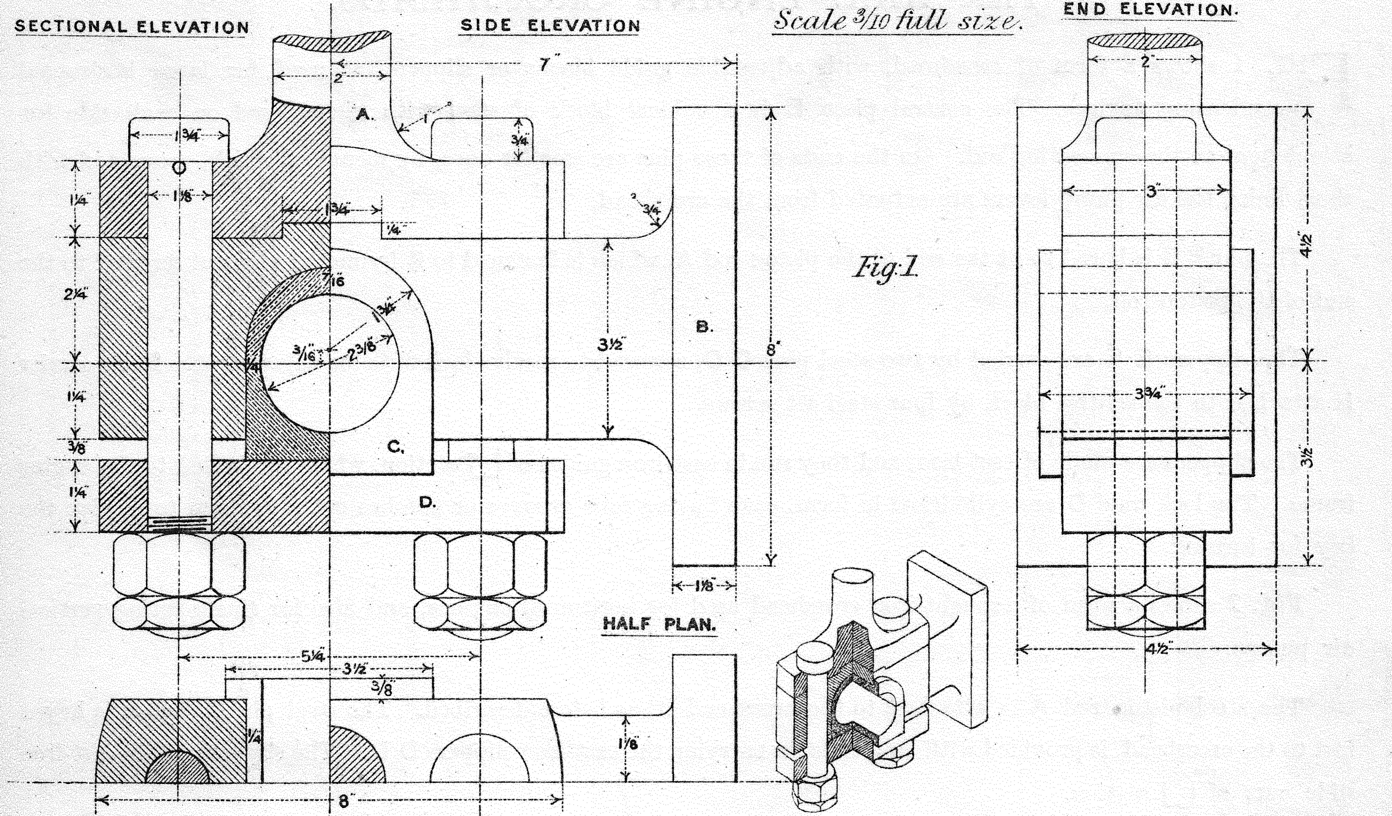


Fig. 1.

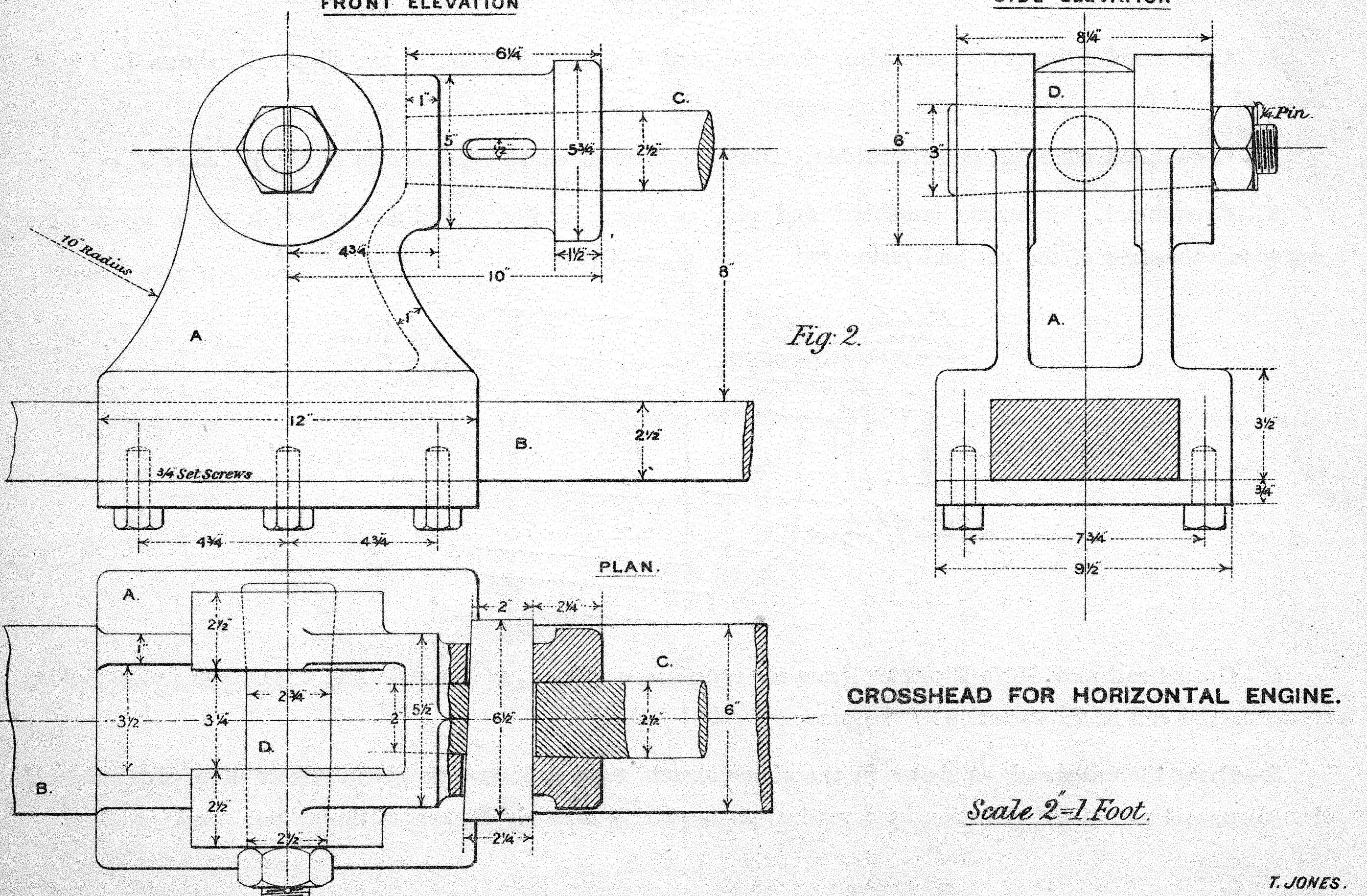


Fig. 2.

CROSSHEAD FOR HORIZONTAL ENGINE.

Scale 2" = 1 Foot.

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Plate XXX.—ENGINE CROSSHEADS.

FIG. 1 shows a form of crosshead, with adjustable guide blocks or slippers, as used for large horizontal condensing engines. The central piece B is a cubical block of steel with pins forged on each side for attachment to the connecting rod. On the ends of these pins are smaller ones, by means of which, connected with short links, the air pump levers are actuated from the crosshead.

The block B is bored to fit the end of the piston rod A, which is fastened to it by means of a nut screwed to the end of the piston rod.

The slippers E E are carried by two steel pins C C, screwed to receive lock nuts for adjustment. Each carrier is attached to the central block by four steel set screws.

The slippers are made of cast-iron, and they run in cast-iron guides of |_| section, which are bolted to the engine frame. The lock nuts D are cylindrical in form, each having four grooves or notches cut into them to receive the key for turning.

Fig. 2 shows a form of wrought-iron crosshead used for horizontal engines, and also for single acting vertical air pumps.

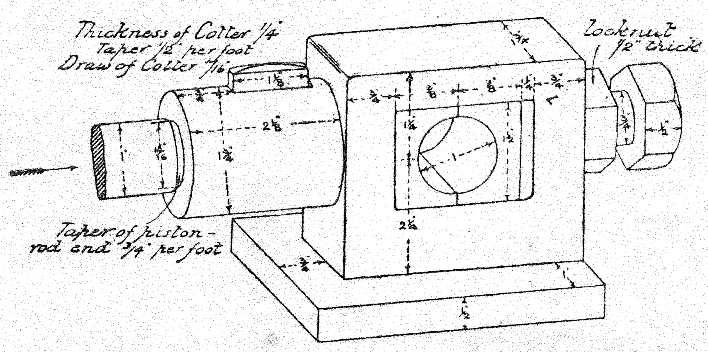
The steel piston rod A is attached to the crosshead B, as before described. The steel pin C, which is keyed fast to the crosshead, is provided with long ends for carrying the cast-iron slippers D D. The slippers run in cast-iron slide bars of |_| section.

EXERCISES.

1.—**Crosshead Slipper.** Draw plan, elevation, and sectional elevation of the slipper E, shown in Fig. 1. *Scale* 6″ = 1 *foot.*

2.—**Crosshead with Adjustable Slides.** Draw the three given views, as shown in Fig. 1. *Scale* 3″ = 1 *foot*

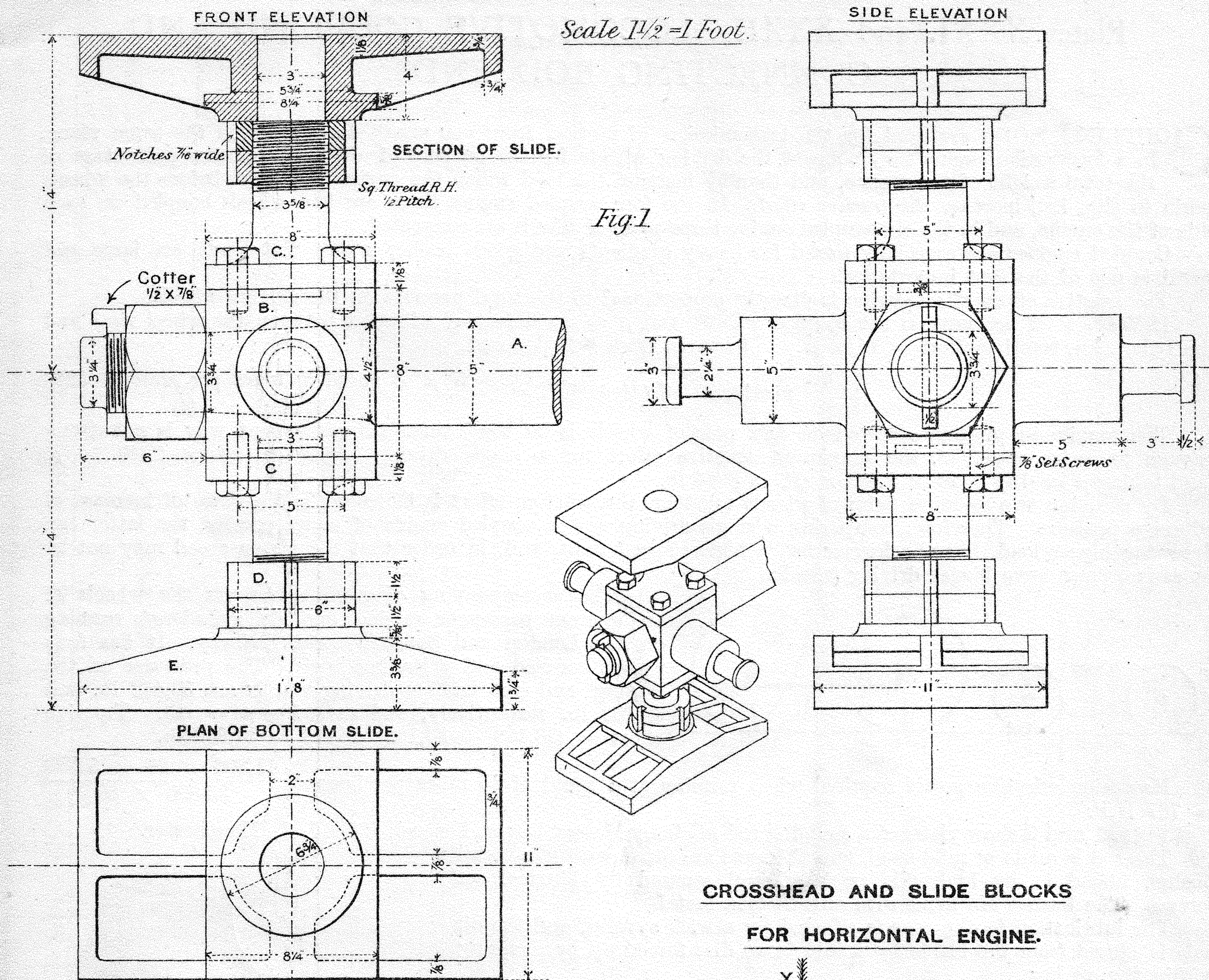
3.—**Crosshead.** Draw the crosshead and pin, as shown in Fig. 2, and add a section made by a plane containing the axes of the pin and piston rod. *Scale* 6″ = 1 *foot.*



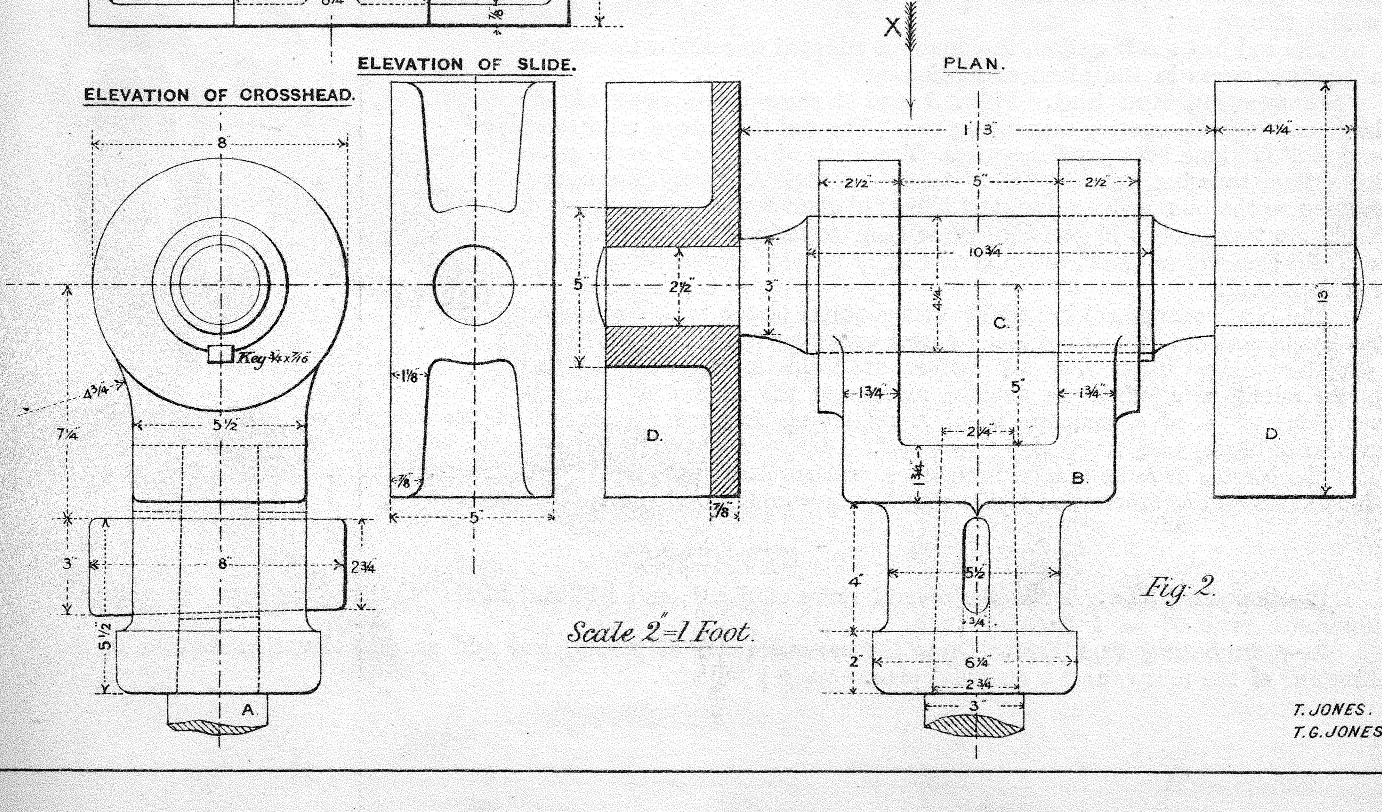
4.—**Crosshead and Slide Blocks.** Draw the complete crosshead, as shown in Fig. 2, and add a view looking on the forked end in the direction of the arrow. *Scale* $\frac{3}{8}$ *full size.*

5.—Draw the crosshead, as shown in the above sketch, to the dimensions given. Show plan, side and end elevations, and a sectional elevation by a vertical plane passing through the axis of the piston. *Scale full size.*

CROSSHEAD WITH ADJUSTABLE SLIDES.



CROSSHEAD AND SLIDE BLOCKS FOR HORIZONTAL ENGINE.



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Plates XXXI. and XXXII.—LOCOMOTIVE COUPLING AND CONNECTING ROD ENDS.

COUPLING RODS are used for the transmission of motion from one crank to another in the same plane. In a locomotive engine they connect the driving wheels for the purpose of utilising a greater percentage of the total weight of the engine, and thereby increase the load which the engine can draw before the wheels begin to slip, i.e., increase the *tractive effort*. In the four-coupled engine there are two wheels coupled on each side of the engine, and in the six-coupled engine three pairs of wheels are connected.

Coupled engines are especially useful for heavy gradients and goods traffic, where the loads are large and speed is not of the first importance.

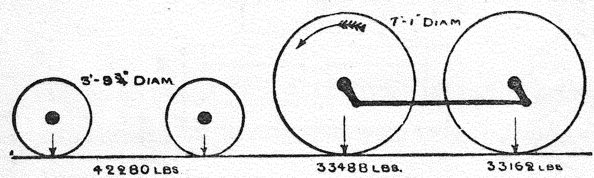
The tractive effort of an engine is directly proportional to the total pressure on the driving wheels.

If W = total pressure on the coupled wheels, and μ = coefficient of adhesion between the wheel rims and the rails, then, tractive effort = $W \times \mu$ (μ varies from $\frac{1}{3}$ to $\frac{1}{10}$), and

$$\text{maximum effective H.P.} = \frac{W\mu \times \text{speed in miles per hour} \times 5280}{33{,}000 \times 60}$$

The maximum pressure allowed on any pair of wheels is 20 tons, since the permanent way is not strong enough to resist more; $\therefore$ the maximum tractive effort for a single driving engine $= 20 \times 2240 \times \mu$, (and taking $\mu = \cdot 18$) $= 20 \times 2240 \times \cdot 18 = 8064$ lb.

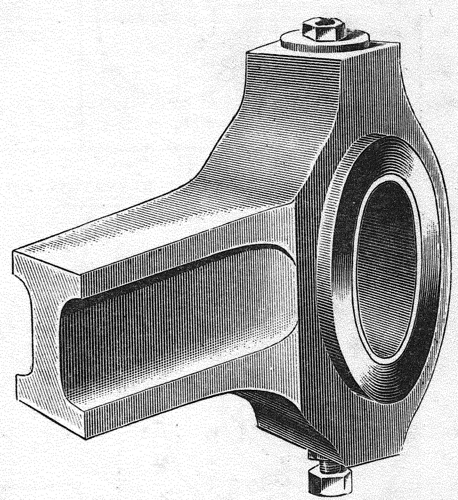
By coupling two or more pairs of wheels together, the tractive effort is increased in the ratio of increase of adhesive pressure. Therefore, comparing a single-driving and a coupled engine of equal powers, the latter can draw the bigger load, but the former has a higher speed limit, and, in order that the piston speed may not be excessive, must have bigger driving wheels.



The accompanying diagram represents the wheels of an express passenger engine (outside cylinders), running on the London and South-Western Railway. It has four wheels coupled and a leading bogie. The pressures on the leading and the trailing driving wheels are 33,488 lb. and 33,162 lb. respectively, and assuming $\mu = \cdot 18$. Tractive effort $= (33{,}488 + 33{,}162) \times \cdot 18 = 11{,}806$ lb.

Maximum effective power required when running at a speed of 25 miles per hour $= \dfrac{11{,}806 \times 25 \times 5{,}280}{33{,}000 \times 60}$ = 787 H.P.

Figs. 1 and 2 show views of a coupling rod made by Messrs. Beyer, Peacock & Co. Ltd., Manchester.



The rod is made of mild steel, and the ends are fitted with gun-metal bushes, forced in by hydraulic pressure, and secured by $\frac{3}{4}''$ steel set screws. The bushes are lined with anti-friction metal.

The lubrication to the pins is effected by a wick syphon, and the oil hole is closed from the inside by a thin brass disc forced up by a spring within the oil box.

The rod has the I section, the one best adapted to resist a thrust and a bending stress in the plane of motion.

Connecting Rod End. Figs. 3 and 4 show three views of the large end of a locomotive connecting rod. The rod is made of mild steel, and is 5′ 11″ long between the centres. The body of the rod is rectangular in section, tapering from $4\frac{1}{2}'' \times 2''$ to $3\frac{1}{4}'' \times 2''$. A wrought-iron strap secured to the butt end by two steel bolts $1\frac{1}{4}''$ diameter at the smaller end, holds the two brasses in position, while their adjustment is effected by a wrought-iron wedge block, which is moved by two $1\frac{1}{8}''$ screws being turned simultaneously.

The block screws are locked by wrought-iron plates, which pass over the heads, and are held in position by steel pins passing through the heads.

The wedge block has an incline of 1 in 8, and the locking plates admit of a minimum angular motion of the screws (11 threads per inch) of $\frac{1}{12}$ of a complete turn, $\therefore$ minimum forward adjustment of the brasses $= \frac{1}{8}$ (minimum vertical motion of block) $= \frac{1}{8} \times \frac{1}{12} \times \frac{1}{11}'' = \frac{1}{1056}''$.

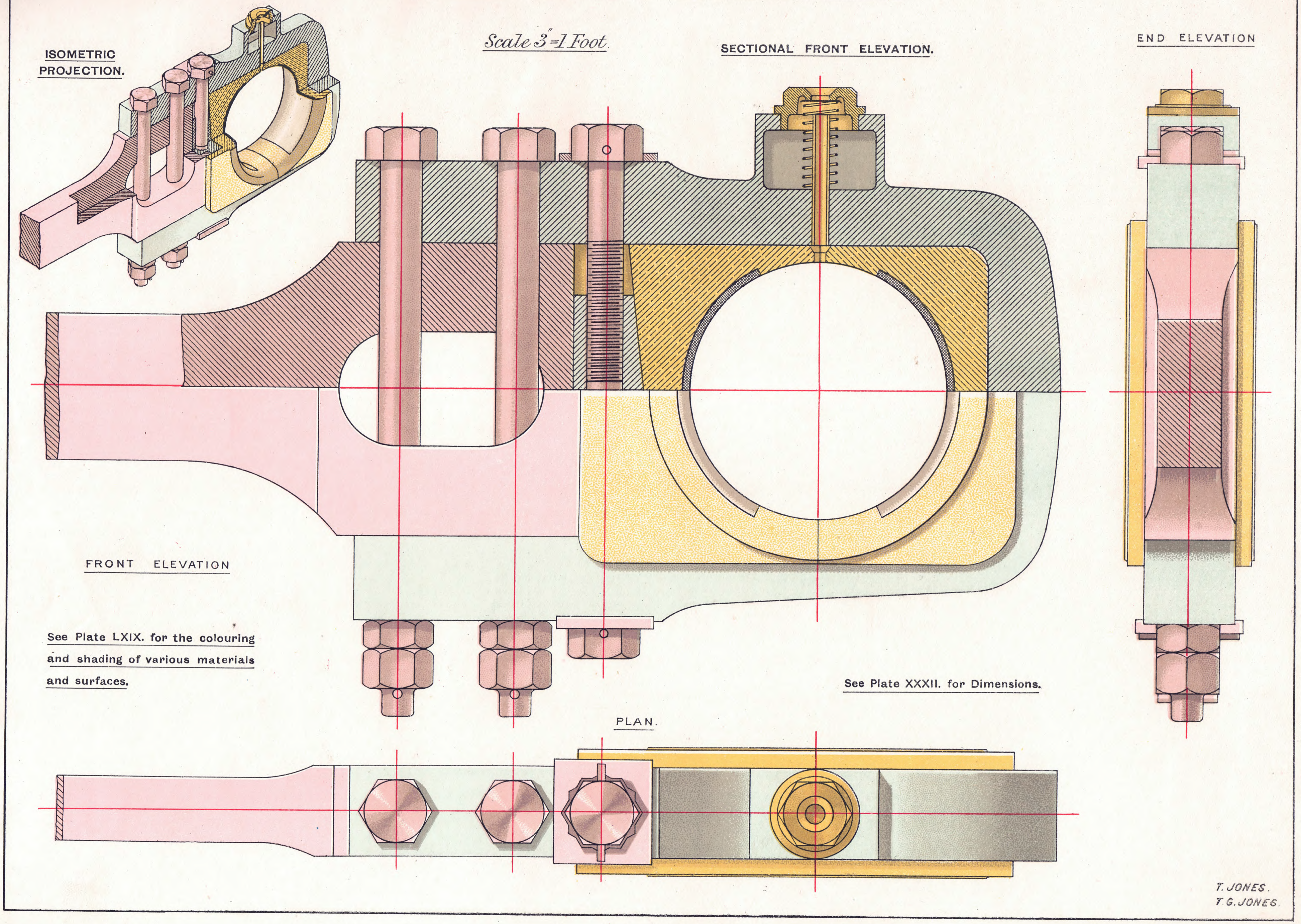
The brasses have flanges on both sides, and are fitted with white metal liners. The oil box is forged on solid, and the method of lubrication is like that for the coupling rod ends.

EXERCISES.

1.—**Coupling Rod.** Draw the several views of Fig. 1, and add an end elevation looking from the centre of the rod. *Scale* 6″ = 1 *foot.*

2.—**Connecting Rod End.** Draw the several views of Fig. 3, and add an end elevation looking in the direction of the arrow, and a sectional plan. *Scale* $\frac{1}{2}$ *full size.*

LARGE END OF LOCOMOTIVE CONNECTING ROD.



LOCOMOTIVE COUPLING AND CONNECTING ROD ENDS.

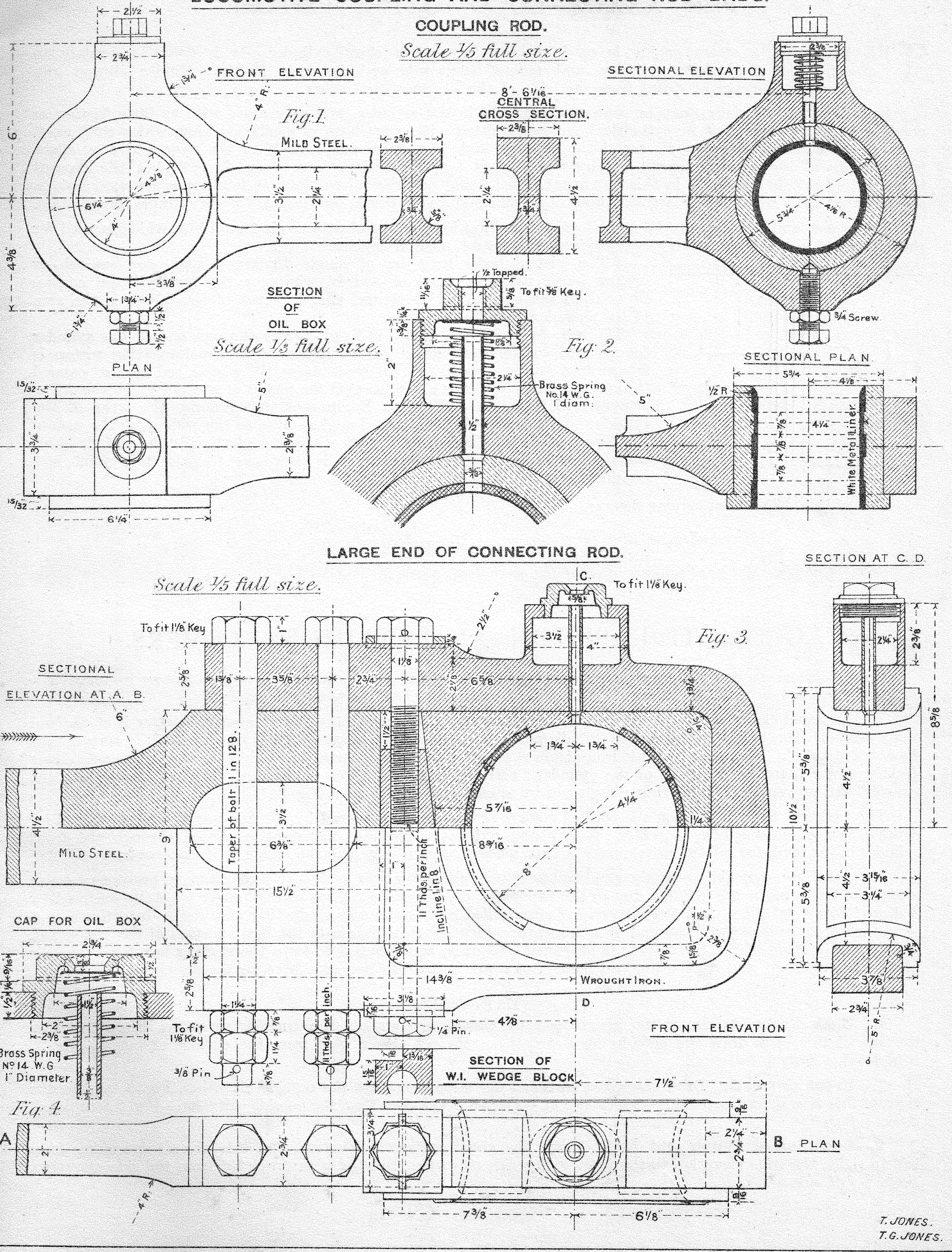


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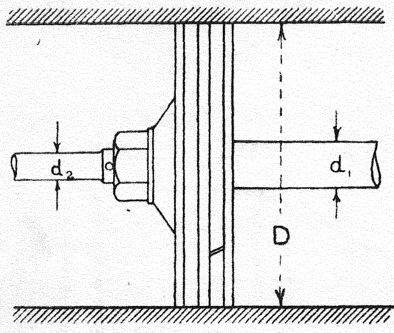
Plate XXXIII.—LOCOMOTIVE PISTON.

THE **piston** of a steam engine is made in the form of a short cylinder. It is fitted accurately to the bore of the engine cylinder, and moves backwards and forwards in the cylinder without any appreciable leakage of steam from one side of the piston to the other.

In order that the piston may be always steam-tight in the cylinder, and may move with as little friction as possible, it is necessary that its cylindrical surface be fitted with some form of elastic packing, which shall exert a small but sufficient pressure all round the cylinder and be able to adapt itself to the wear and any slight irregularities of the cylinder barrel. Many complicated piston packings have been devised, but the present tendency is towards simplicity in this important detail; and not more than three or four types are used now to any large extent. Even for the largest pistons of marine engines the piston packings are of very simple construction.

The oldest and most widely adopted form of packing is the **Ramsbottom ring packing,** which is used for the pistons of locomotives, small steam engines, gas, petrol, and oil engines.



The piston body may be of single thickness—plane or conical—or hollow block pattern; but, of course, whatever the form of the surface exposed to the steam, the maximum effective area of the piston is the area of the cylinder barrel.

The rod, to which the piston is fixed, passes through one of the cylinder covers, and so the work done by the steam on the reciprocating piston is transmitted to the crosshead. In some cases the piston rod is also carried through the other cylinder cover, so that the piston of another cylinder may be attached to it—as in a Tandem Engine—or, to minimise the wear of the cylinder—in the case of a horizontal engine—due to the weight of the piston.

The piston rod diminishes the effective area of the piston, and, in most cases of estimation of horse-power, it is necessary to take this point into consideration.

Referring to the accompanying figure:

$$\text{Mean effective area of piston} = \frac{\pi}{4}\left(D^2 - \frac{d_1^2 + d_2^2}{2}\right)$$

NOTE.—*Calculate the mean effective area of the given loco. piston.*

LOCOMOTIVE PISTON AND CYLINDER COVERS.

On the accompanying plate are given drawings of the piston and cylinder covers of a heavy locomotive—outside cylinders—as made by Messrs. Hawthorne, Leslie & Co. Ltd., Newcastle-upon-Tyne.

The sectional elevation, Fig. 2, shows each cylinder cover relative to the piston when the latter is at the end of its stroke. The cylinder is bored to 20″ diameter, and the piston body, which is of cast-steel, is turned to a diameter of $19\frac{15}{16}$″. There are two cast-iron piston rings, 1″ × $\frac{1}{2}$″ rectangular section, which are turned to a diameter slightly larger than the diameter of the cylinder; and, when a short piece is cut out, they are sprung into the grooves, and prevented from moving along them by screws as shown at X in Fig. 3.

The piston is fixed tightly on the conical portion of the steel piston rod by the gun-metal nut (see **Fig.** 4 for details), which is locked by a cotter. The piston rod—at a reduced diameter—is carried through the front cylinder cover, and, for safety, moves inside a tube whose flange is bolted to the flange F of that cover.

Each cast-iron cylinder cover is secured to one of the cylinder flanges by 20—1″ studs; and, to prevent excessive steam condensation in the cylinder due to the exposed ends, the outer surface of each cover is totally enclosed by a dished cover $\frac{3}{16}$″ thick, and the intervening space filled with some non-conducting material. Into the boss C of the front cover is screwed the end of a lubricator which serves to lubricate the front portion of the piston rod. The central portion of the back cover is projected outwards, and recessed to form the stuffing box for a metallic piston-rod packing; and, to the top and bottom of it are bolted the two slide bars for the crosshead.

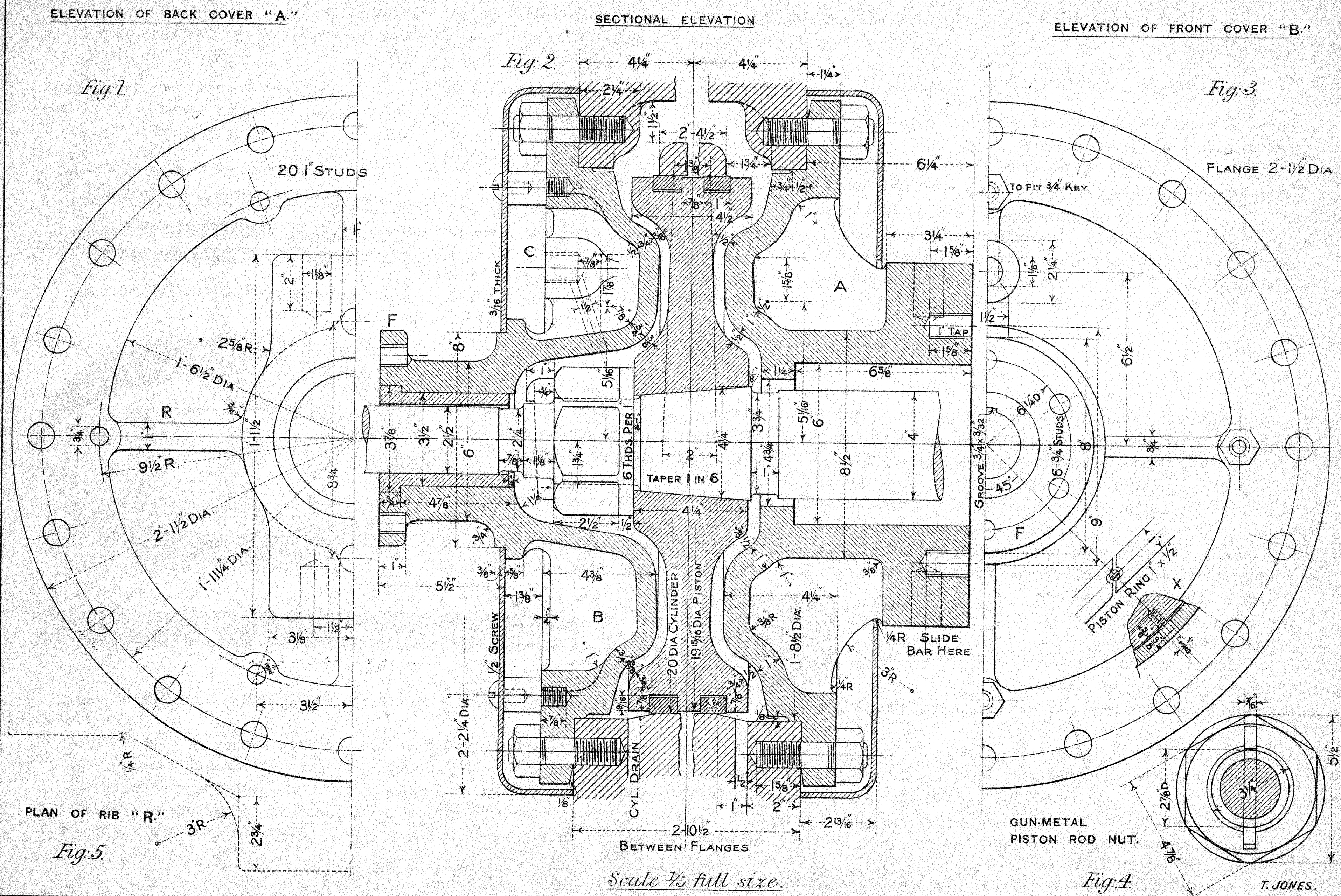
EXERCISES.

1.—**Back Cover A.** Draw the given sectional elevation, complete the outside elevation, Fig. 1, and to the left of this view draw the outside edge elevation. Project also a plan from the outside elevation—**Fig.** 1, completed. *Scale $\frac{1}{3}$ full size.*

2.—**Front Cover B.** Draw the given sectional elevation, complete the outside elevation, Fig. 3, and to the right of this view draw the outside edge elevation. Project also a plan from the outside elevation—Fig. 3 completed. *Scale $\frac{1}{3}$ full size.*

3.—**Locomotive Piston and Rod.** Draw the given sectional elevation, the side elevation looking from left to right, the edge elevation to the right of the side elevation, and the plan projected from the side elevation. *Scale $\frac{3}{8}$ full size.*

LOCOMOTIVE PISTON AND CYLINDER COVERS.



Scale ⅓ full size.

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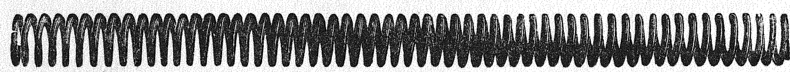
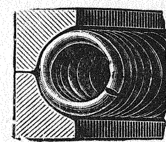
Plate XXXIV.—36″ PISTON: PISTON VALVE.

PISTON: The cast-iron body of this piston is made hollow, and the flat faces are stiffened inside by ten thin webs. The steel piston rod is secured to the piston by a nut which is locked by means of a steel cotter. In order that the back cylinder cover need not be dished excessively on account of the projecting nut, the lower portion of the nut is turned cylindrical and fitted into the boss of the piston.

This piston is for the low-pressure cylinder of a compound condensing engine, so the rod is carried through the back cover for connection with the air pump piston. In this arrangement the weight of the piston is partially carried by the rod, and undue wear on the lower side of the cylinder is thus prevented.

The packing shown here is the "Lancaster" packing, and consists of a straight helical spring bent into a circular form and then compressed in length to fit into cast-iron packing rings, or casings D D (*see sketches*). These casings are turned a little larger in diameter than the cylinder barrel; short lengths are cut out of them so that they may be compressed into the cylinder. The expansive tendency of the rings is increased by the coil inside, which also tends to assume its original form and length, and consequently exerts an outward pressure on the rings.





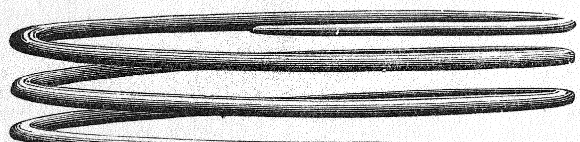
SECTION OF RING AND SPRING SHOWING HOW THE SPRING BEDS TO WHOLE OF THE INNER SURFACE OF RING.

Piston Valve.—The valve consists of two small pistons, P_1P_2, separated by a hollow distance piece D, and secured by a nut and taper pin to the shouldered valve spindle. This form of valve differs from an ordinary flat slide valve in that the working face is cylindrical instead of plane.

By this modification the valve is balanced, there being no pressure of the valve against the cylinder face due to the steam. It is therefore much used for the high-pressure cylinders of compound and triple expansion engines, particularly marine engines.

The piston ends of the valve, slide over oblique slots or ports extending round the cast-iron or steel liners in which they work. The slots, being in connection with the steam passages to the cylinder, form the steam ports.

In order that the valve may work steam tight in the liners each end must be fitted with some simple form of piston packing, which is capable of resisting the collapsing steam pressure on the outer circumference due to the working of the valve over the ports, and, at the same time, only exert a gentle pressure to compensate for wear of the rubbing surfaces. The valve shown is for a horizontal engine, and is fitted with the "Lancaster" Serpent Coil, which is a short helical spring of stout steel wire. (*See accompanying perspective illustration.*)



The springs are enclosed in ordinary split casing rings, and by compressing them to bring the rings together, they increase in diameter, and so exert an outward pressure on the rings.

The oblique slots in the rings are closed by small brass tongues or making-up pieces. The face of each piston is the same as the length of the face of the common valve, the inside and outside laps being also the same. The admission of steam to the cylinder is regulated by the two outer ends of the valve, and the steam exhausts into the space between the two pistons.

EXERCISES.

1.—**36″ Piston.** Draw the several views of the piston, completing the plan. *Scale* 4″ = 1 *foot.*

2.—**Piston Valve.** Draw the given plan of the valve showing the full length, and add an end view looking on the nut end of the rod. *Scale* $\frac{3}{4}$ *full size.*

QUESTIONS.

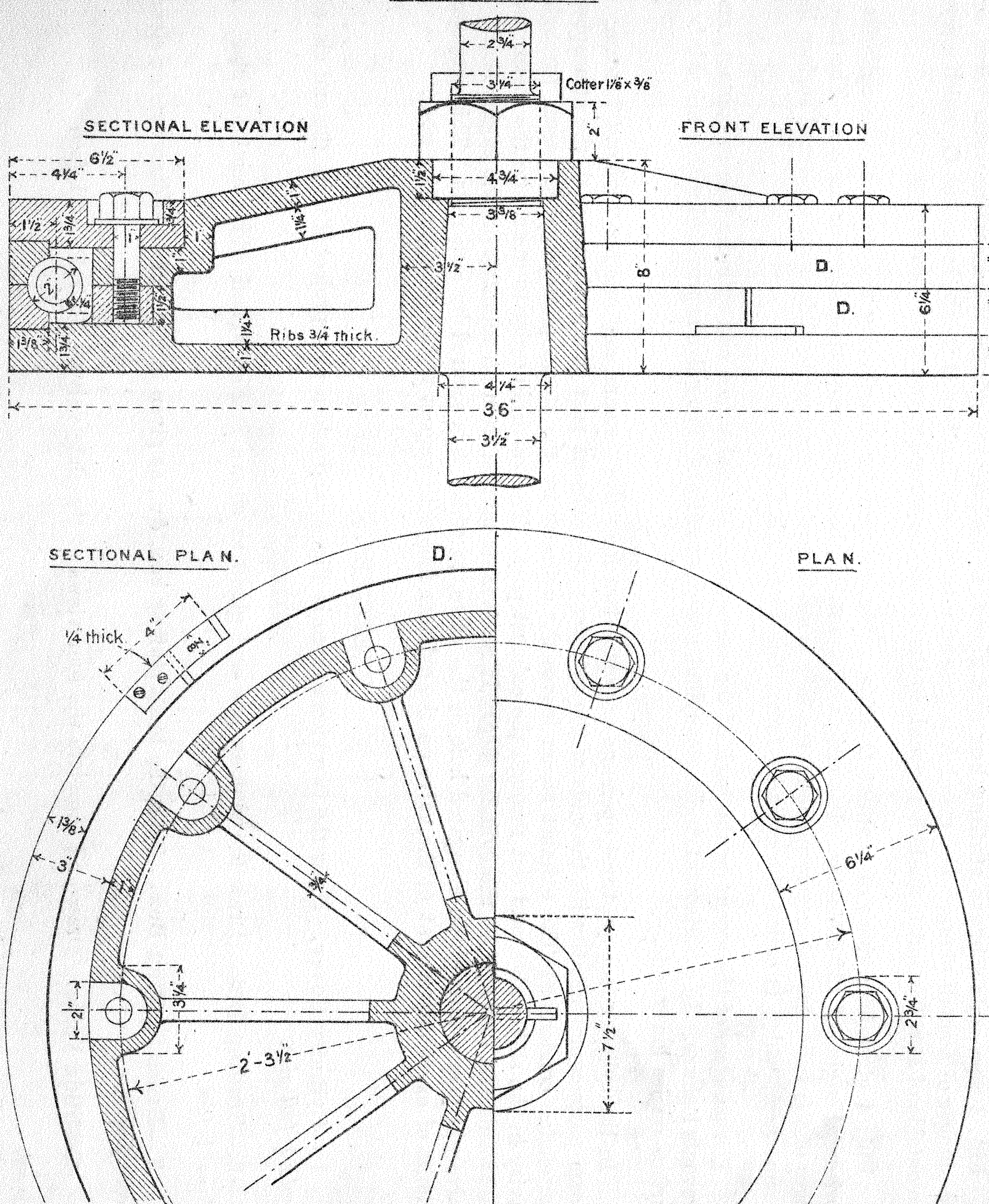
1.—If the casings of a 36″ piston are turned $36\frac{1}{2}$″ diameter, find the amount to be cut out so that it may fit the cylinder.

2.—Explain the advantage of the Junk ring screws, fitting into brass nuts instead of into the body of the piston.

3.—Distinguish between a piston, a plunger, and a pump bucket. Give a sketch of each.

36" PISTON.

Scale 1¾" = 1 Foot.



PISTON VALVE.

Scale ½ full size.

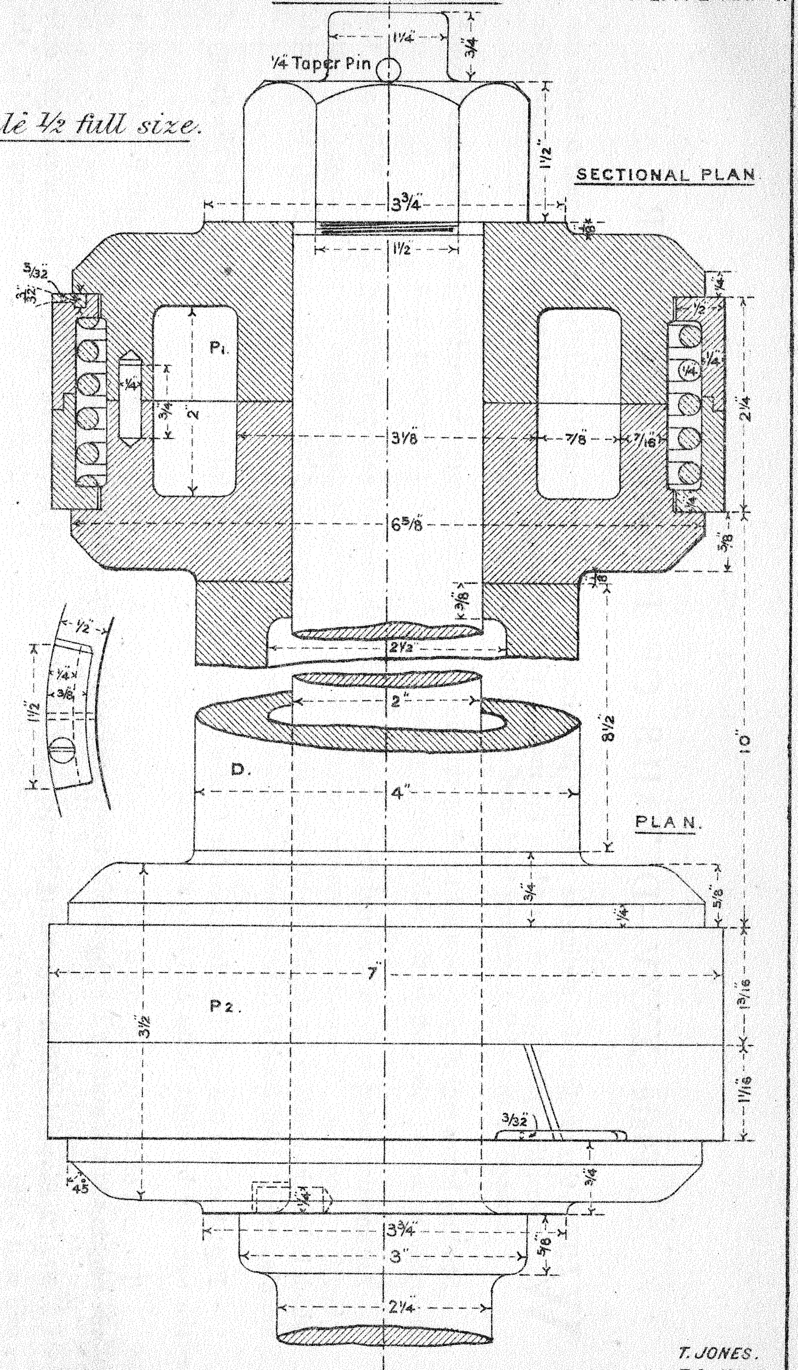


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Plate XXXV.—DETAILS OF ENGINE CYLINDER.

WHEN the motion of a sliding or rotating piece inside a vessel is transmitted to parts external to it or *vice versa*, the cylindrical rod, to which the moving piece is attached, must pass through the vessel without the leakage of any of the working fluid, and also with very little friction.

Stuffing boxes and glands are used to effect this end, as in the steam engine where the piston rod passes through the cylinder cover ; also, where the valve spindle passes through the end of the steam chest.

An annular space is formed round the rod, and filled with rings of asbestos packing, such as Bell's and the "Codifex" packing. The packing is pressed into the cavity or stuffing box and against the rod by means of a gland, which is held in position by stud bolts attached to the flange of the stuffing box. Metallic packings of various kinds are also used. (*See Book III. for drawings of metallic packings.*)

Fig. 1 shows the cast-iron **front cover** C of a cylinder for a horizontal engine. It is secured to the flange of the cylinder by eight bolts and two stud bolts.

The end of the stuffing box B, and the whole of the gland A, are lined with brass. The gland is forced against the packing in the stuffing box by the nuts screwed on the three stud bolts.

Fig. 2 shows the form of **stuffing box** B, and gland A, as used for a valve spindle. In this example only two stud bolts are necessary for the gland, and the flange is shaped to suit these. The parts through which the spindle passes are brass bushed, as in Fig. 1.

Fig. 3 represents a **brass stuffing box and gland,** as used for the spindle of a steam or water valve, or for the plunger of a small pump. The gland A is pressed into the stuffing box B by the brass nut C, which screws on the outside of B. The nut is made hexagonal, so that it can be turned round with a screw key.

Fig. 4 gives four views of a **short D slide valve,** as used for distributing the steam to the cylinder of a horizontal engine of 12″ diameter. It consists of a cast-iron box, with projecting edges or flanges. The face and edges of the valve are accurately planed ; the face is made smooth and true to slide upon the trued surface in the steam chest, which is called the cylinder face. See Plate XXXVI.

The pressure of the steam on the back of the valve keeps the valve face and cylinder face in contact, and so prevents leakage. In a horizontal engine the cylinder face is vertical. The valve is provided with a suitable bearing surface on its lower side.

In the cylinder face are three openings or ports—two for "*live*" steam, and one for "*exhaust*" steam. The steam ports communicate with the ends of the cylinder ; the exhaust port communicates with the atmosphere or the engine condenser. The exhaust steam leaves the cylinder by the steam passages, then passes through the cavity in the slide valve, and out through the exhaust port. The valve, in order to distribute the steam, travels backwards and forwards on the cylinder face. It receives its motion from an eccentric on the engine shaft. The valve rod passes through an elongated hole in the valve, to which it is secured by lock nuts. This arrangement admits of exact adjustment of the valve on the cylinder face with respect to the ports, and for wear.

EXERCISES.

1.—**Cylinder Cover.** Draw the two views given in Fig. 1, and add a plan projected from the end elevation. *Scale ½ full size.*

2.—**Stuffing Box.** Draw the stuffing box and gland as shown in Fig. 2, and add a sectional plan. *Scale* 9″ = 1 *foot.*

3.—**Brass Stuffing Box.** Draw the brass stuffing box, &c., as given in Fig. 3, and add an outside elevation. *Scale full size.*

4.—**Slide Valve.** Draw and complete the four views of the slide valve. *Scale* 9″ = 1 *foot.*

DETAILS OF ENGINE CYLINDER.

CYLINDER COVER WITH STUFFING BOX.

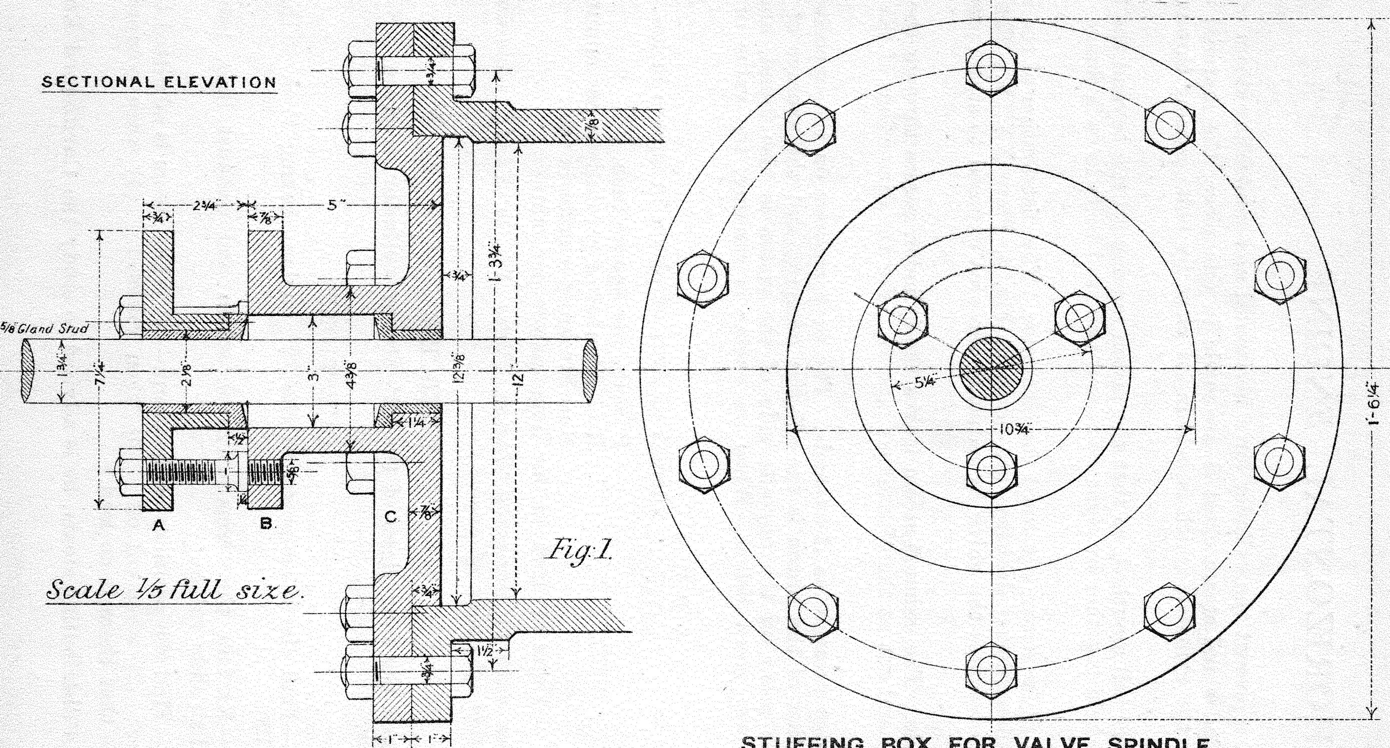

Fig: 1.

STUFFING BOX FOR VALVE SPINDLE.

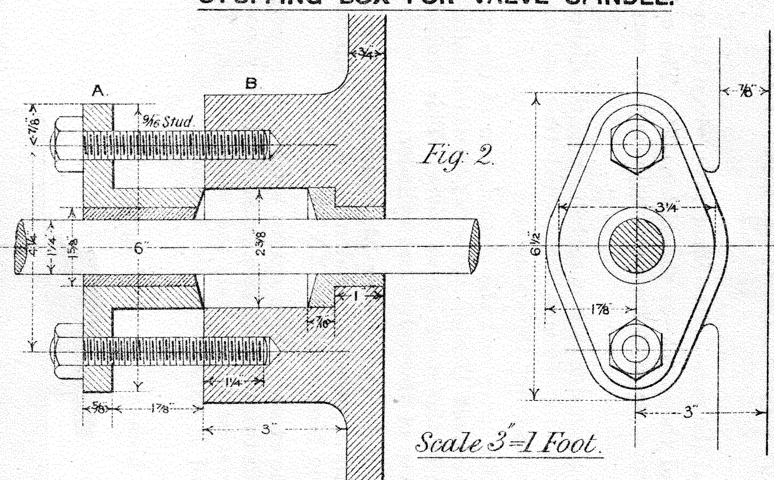

Fig: 2.

BRASS STUFFING BOX.

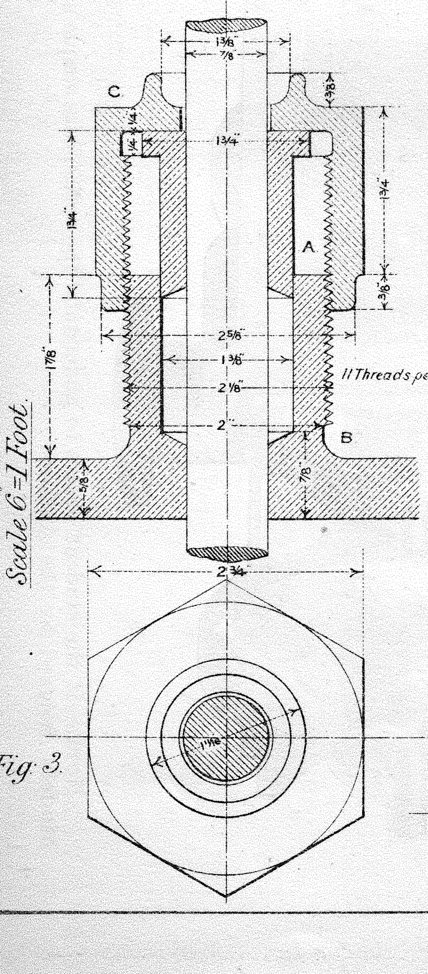

Fig: 3.

SLIDE VALVE AND SPINDLE.

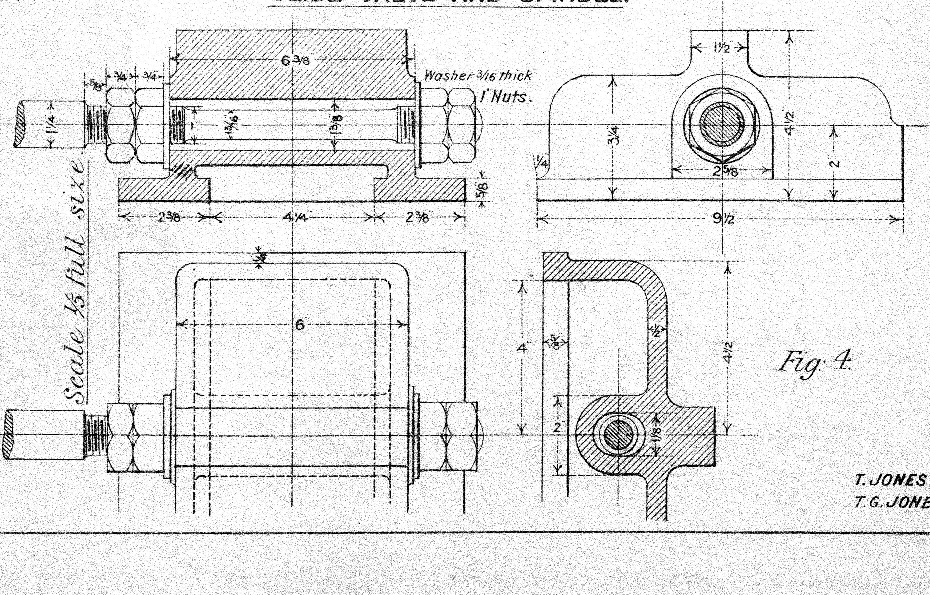

Fig: 4.

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Plate XXXVI.—CYLINDER FOR HORIZONTAL ENGINE.

THE drawings on the opposite page and the perspective illustration given below, represent the cylinder for a horizontal non-condensing steam engine.

The cylinder is 12″ diameter and the stroke of the piston 19″. The cylinder and steam chest S are made in one piece, the material being tough close grained cast iron. Facings are cast on the engine bed to carry the cylinder, which is held in its place by bolts passing through the engine bed and the feet on the cylinder.

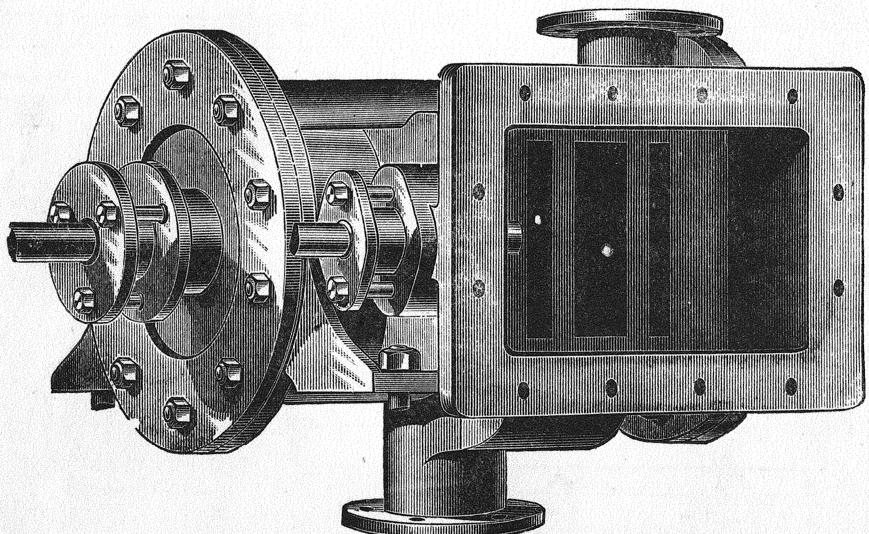
The cylinder is accurately bored to fit the piston, the ends being slightly dished or bell mouthed, so that the piston moving a little beyond the bored part wears the surface evenly. The covers and flanges are turned true and are held together by eight bolts and two stud bolts in each flange. The stud bolts are used where the steam passages enter the cylinder.

The small distance between the cylinder cover and the piston when at the end of its stroke is termed the clearance. The distribution of the steam to the cylinder is effected by means of the short D slide valve V, of which fully dimensioned drawings are given in Fig. 4, Plate XXXV. The back of the valve V fits against the inner face of the steam chest cover. This *partially balances* the valve by reducing the area exposed to steam pressure, and therefore the friction between the faces of the valve and cylinder.

Steam is admitted to the steam chest S by the 3″ pipe M, cast on the top side. The exhaust steam leaves the cylinder, passes through the slide valve V, through the exhaust port, and then round the underside of the cylinder, and out at the 4″ exhaust pipe N. For the dimensions of the front cover, slide valve, and valve spindle stuffing box, see Plate XXXV.

The piston P is moved forward by the steam, which exerts a continually *varying* pressure as it expands. The *back pressure* on the opposite side of the piston also varies. The difference between these two pressures in any position of the piston is the *effective* pressure, and the average of these effective pressures is the *mean effective pressure* throughout the whole stroke. The *mean* on the opposite side of the piston is also found, and the average of these two means is the value of P required in the formula for horse-power given below. When an engine is running, the value of P is obtained from indicator diagrams.

To find the horse-power H of an engine :—



$$H = \frac{2\,A\,P\,R\,S}{33{,}000}$$ where A = Area of piston in square inches.

P = lb. per square inch.
R = Number of revolutions per minute.
S = Stroke of engine in feet.
33,000 = Units of work in one horse-power per minute.

(*See notes to Horizontal Engine, Plates XL. to XLIII. inclusive, for calculation of power of engine.*)

EXERCISE.

Draw the four views of the 12″ **steam cylinder** as given. *Scale* $\frac{1}{4}$ *full size.*

QUESTIONS.

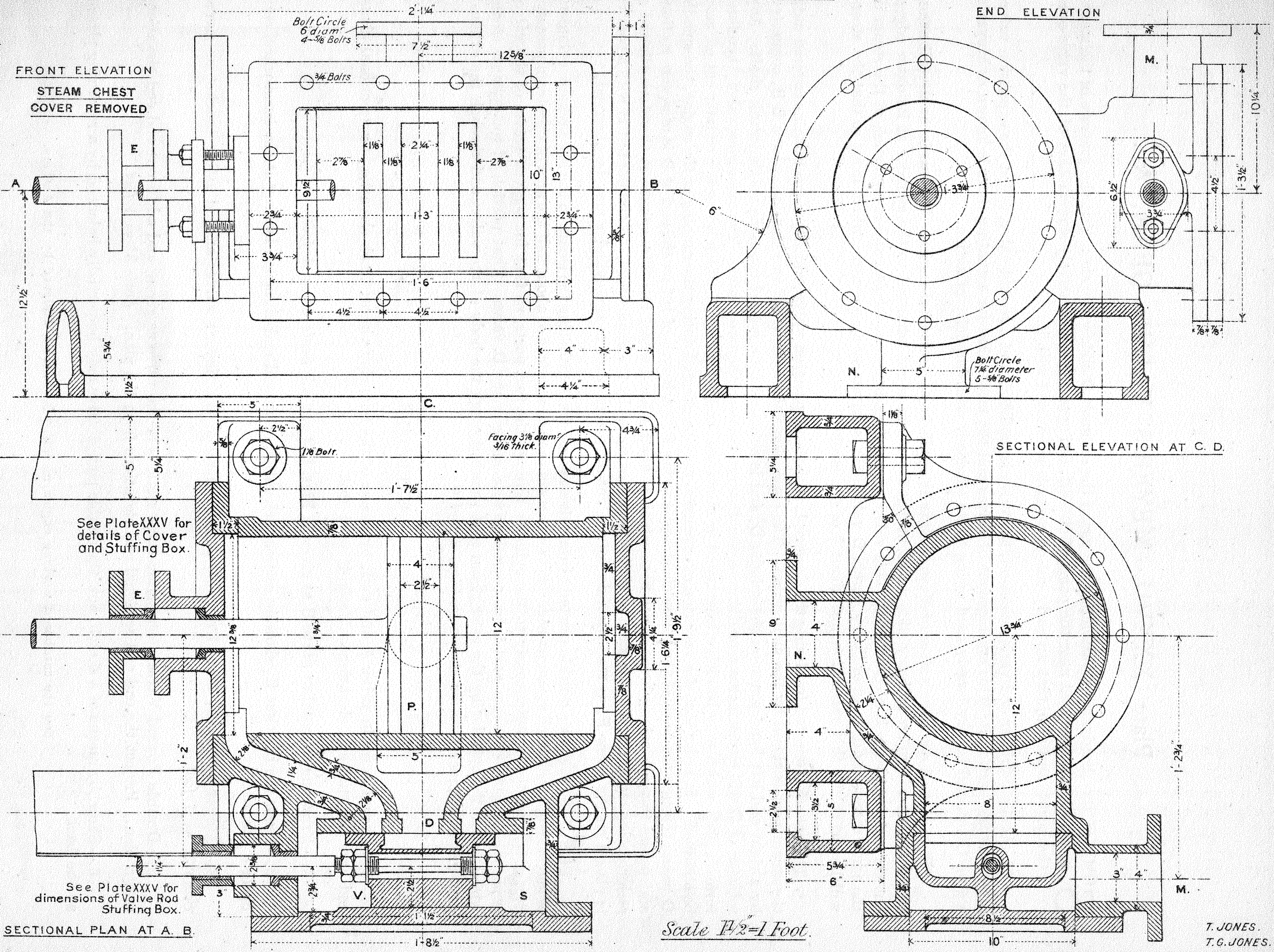
1.—Find the horse-power of the 12″ engine, taking P = 40 lb. and R = 120.

2.—Explain the terms : Travel, steam and exhaust laps, lead, as applied to slide valves.

3.—Find the positions of the crank—(1) When steam is admitted to the cylinder ; (2) when cut off ; (3) when released ; (4) when compressed. Assume that there is no lead.

NOTE.—If the student wishes to understand the action of a simple slide valve in distributing steam, he is advised to study the **Cardboard Engine Models** designed by the Authors.

CYLINDER FOR HORIZONTAL ENGINE.



Scale $1\frac{1}{2}''$=1 Foot.

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Plate XXXVII.—MEYER'S VALVE GEAR, &c.

SOME of the disadvantages attending the use of the common D slide valve are: that the pressure of the steam is considerably reduced near the point of **cut-off,** owing to the comparatively slow motion of the valve over the port; and also, that an early cut-off cannot be effected with it.

To obtain a quicker and variable cut-off, the valve is modified, and a second valve, called the expansion valve, placed on the back of it. This arrangement of valves and the mechanism for actuating them is known as **Meyer's Valve Gear,** and is very largely used owing to its efficiency and simplicity of action.

***Figs. 1 to 4** show the valves of Meyer's Gear as applied to a horizontal engine. The **main or distribution valve** A differs from an ordinary slide valve, in that the steam is not admitted to the cylinder at its outer edges, but through a passage L at each end. This valve effects the admission, release, and compression of the steam in the same manner as does an ordinary simple D valve. It is driven by an eccentric, which is set so that the steam will be cut off by this valve alone, at about $\frac{3}{4}$ of the stroke.

The cut-off is effected by closing the passage L with the **expansion valve,** which passage remains covered until the steam port is closed by the main valve. The expansion valve consists of two cast-iron plates B B, connected by a right and left hand screw on the spindle D, and is actuated by a second eccentric on the crank shaft. Any rotation of D will either increase or diminish the distance between the two plates. The expansion valve, therefore, forms a plate which may be either lengthened or shortened, and by means of which an earlier or later cut-off may be obtained. In order that the variation of expansion may be effected during the running of the engine, the spindle D passes through the back of the steam chest and a fixing E. A length of 10″ on the end of D is of square section, and passes through a long brass screw F to which is fixed a hand-wheel H. The spindle is therefore allowed a reciprocating motion, and at the same time may be turned round by the hand-wheel. A circular nut H with a pointer is placed on the screw F, so that the value of the cut-off taking place in the cylinder may be observed.

Figs. 7, 8, 9, show a **double ported slide valve** as used for distributing the steam to the low pressure cylinders of marine engines. Each steam passage has two ports in the cylinder face of about half the area of the steam passage. Steam is admitted simultaneously to the cylinder through the two ports, so that for a given travel of the valve, twice the area of the steam port is uncovered as with an ordinary D valve. That portion of the exhaust steam which leaves the cylinder by the *outer* steam port passes through the body of the valve, and through the exhaust port in the cylinder face, as shown in Figs. 8 and 9.

EXERCISES.

1.—Show the **main valve A and rod,** by means of the following views: Back elevation, two sectional plans, end and sectional end elevations. *Scale* 4″ = 1 *foot.*

2.—Draw one **expansion plate** B as given in Figs. 1, 3 and 4. *Scale* 6″ = 1 *foot.*

3.—Draw the **bracket** E, showing sectional elevation, plan and end elevation looking at the hand-wheel end. *Scale* $\frac{1}{2}$ *full size.*

4.—**Meyer's Valves, &c.** Draw the various views of the Meyer's valves and adjusting bracket as given. *Scale* 3″ = 1 *foot.*

5.—**Double Ported Valve.** Draw the three sections of the valve as given, and add front and end elevations and a plan. Project the end and sectional end elevations from the plan. *Scale* 3″ = 1 *foot.*

6.—Show, by means of a sketch, how the double ported valve may be partially balanced by a relief ring placed at the back.

* See the cardboard working engine models, "X" Series, designed by the Authors.

MEYER'S VALVE GEAR FOR HORIZONTAL ENGINE.

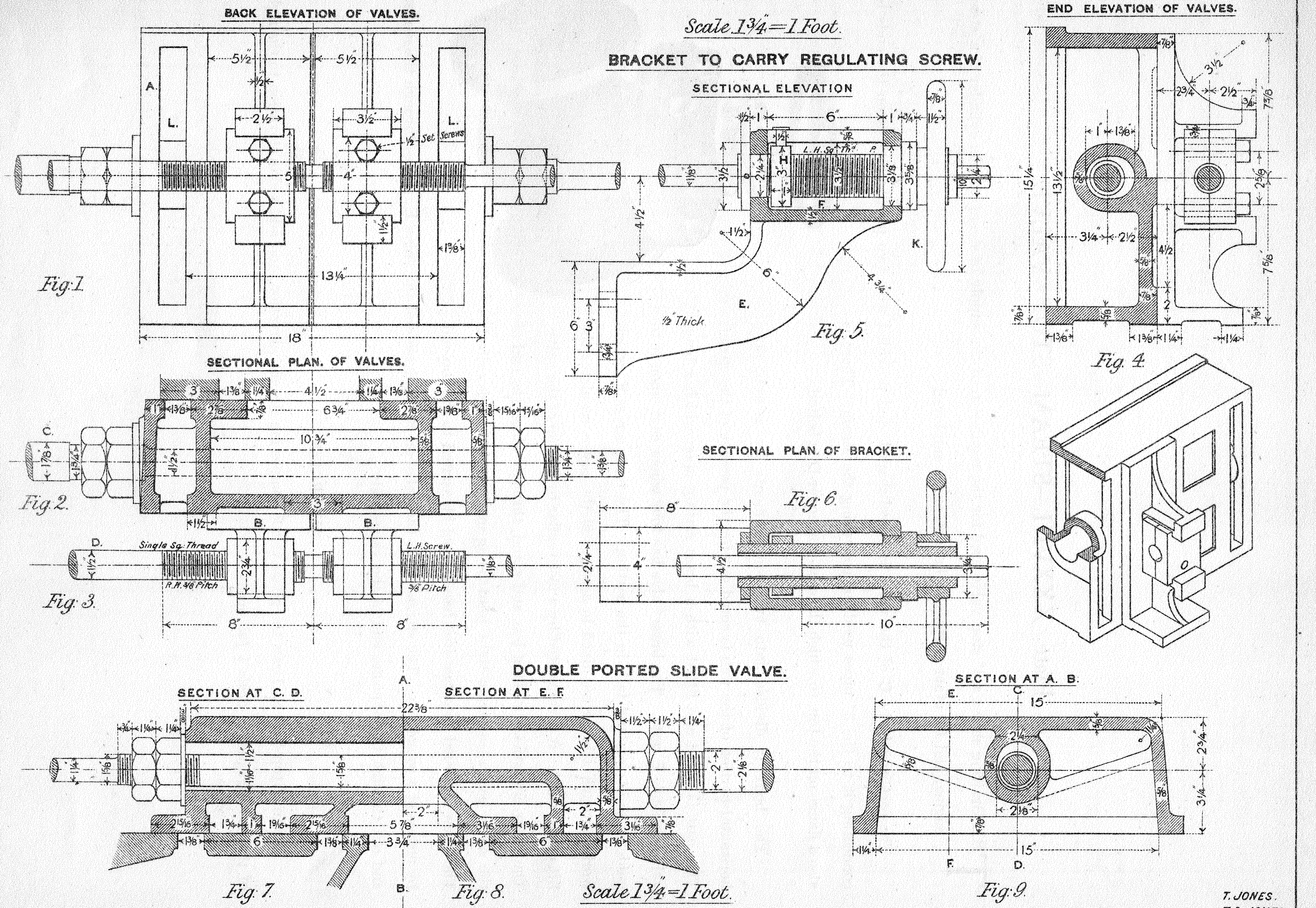

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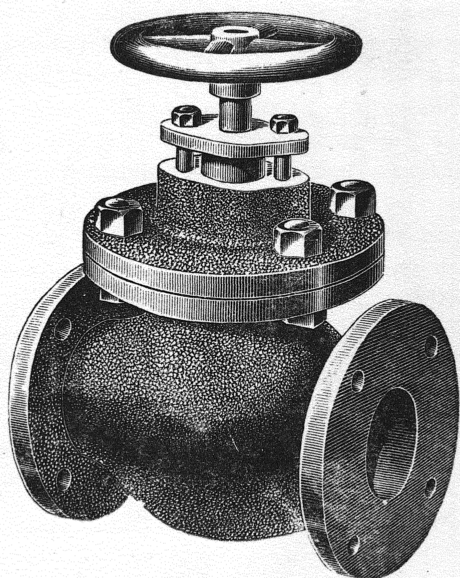
Plate XXXVIII.—STEAM STOP VALVE.

THE Drawing shows a 3″ stop valve used for the regulation of the supply of steam to the cylinder of a steam engine, and for other purposes where steam is employed.

The cast-iron body C of the valve is barrel-shaped, and provided with two parallel flanges on the ends for connecting it to the 3″ pipes used for conveying the steam. There is also a flange on the top, to which the cast-iron cover D is bolted.

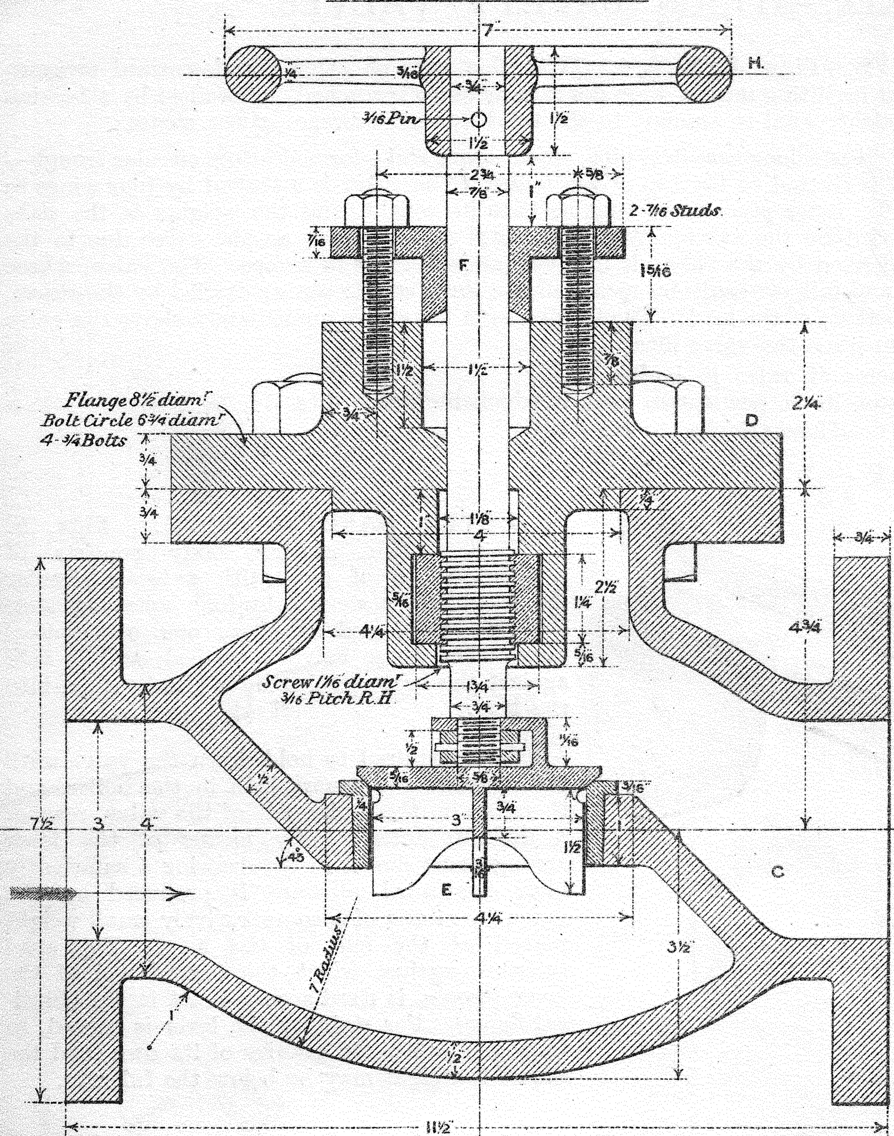


The body is divided into two compartments by a web. The horizontal portion of the web is cored out for the passage of the steam. This part is bored and fitted with a brass seating upon which a brass valve or clack E rests. The brass valve has four wings cast upon it to guide it as it rises and falls in the seating. The brass screw F, which works through a brass nut fixed in a recess in the cover, regulates the lift of the valve : it is connected by a loose collar to the top of the valve. The cover D carries a stuffing box and gland, through which the end of the screw passes—a steam tight joint being made as before described. The screw is turned by a cast-iron hand-wheel H secured to the top end by means of a round pin. The steam enters the valve on the underside of the valve E, passes through the seating and out through the opposite end of the valve body.

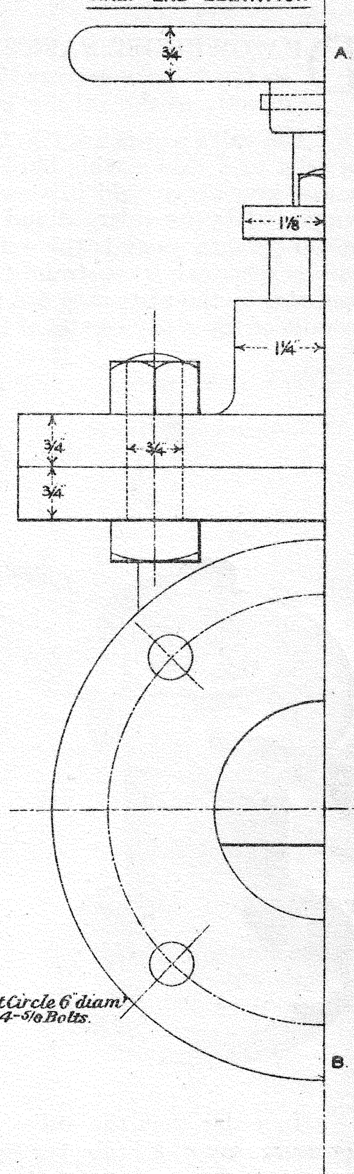
EXERCISE.

Steam Stop Valve. Draw the given views, completing the end elevation, and add a side elevation to the right of the end elevation. *Scale ½ full size.*

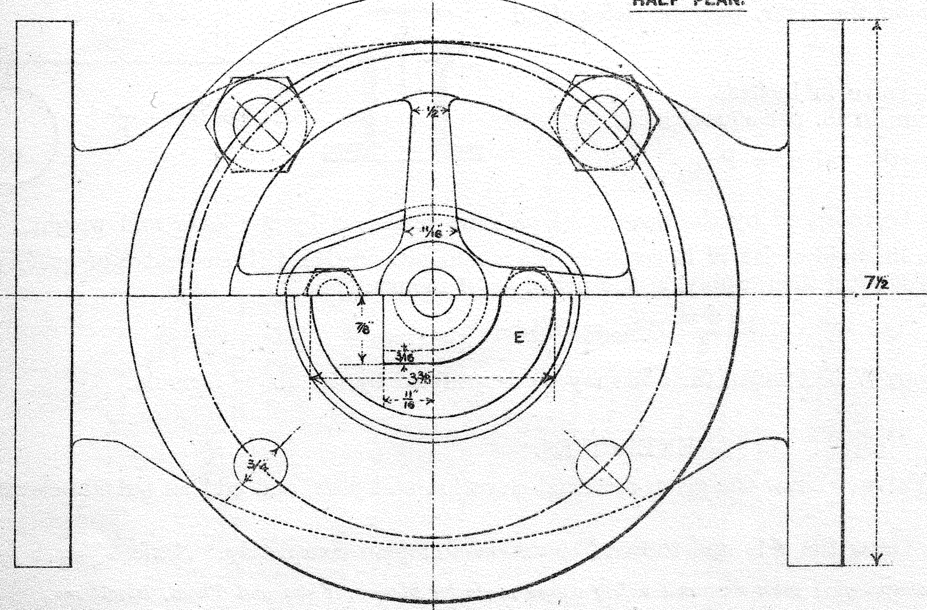
SECTIONAL ELEVATION AT A. B.



HALF END ELEVATION



HALF PLAN.



HALF PLAN. | COVER REMOVED.

STEAM STOP VALVE.

FOR 3" PIPE.

Scale 3/8 full size.

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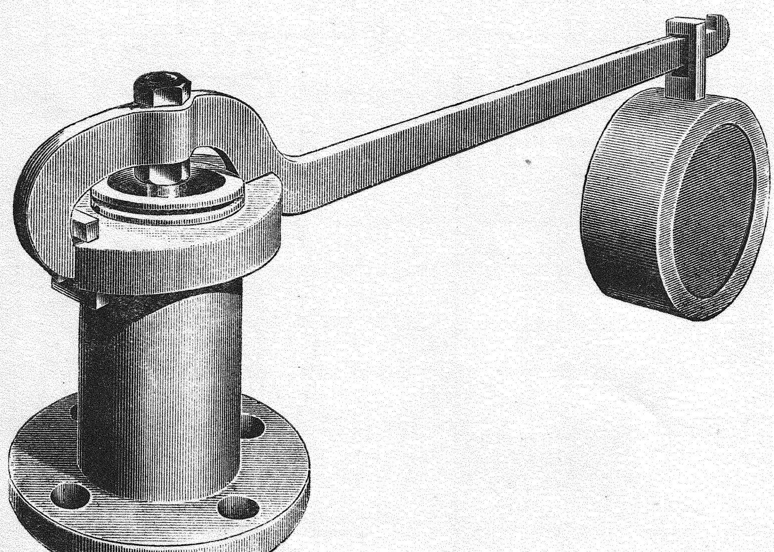
Plate XXXIX.—STEAM SAFETY VALVES.*

DEAD-WEIGHT SAFETY VALVE. Figs. 1, 2, 3, 4. With this type of valve the downward pressure on the valve, necessary to prevent its lifting until a desired steam pressure is reached, is obtained by attaching directly to the valve a heavy weight equal in amount to the total upward pressure of the steam.

The valve seat is fixed on the top end of a long cast-iron pipe, whose lower end—formed into a circular trough—is fixed to a steel seating block, which is riveted to the top of the boiler. The valve is attached by four wings to a cast-iron sleeve, which passes over the inner pipe, and hangs so that its weight, and the weights of the slabs supported by its enlarged end are carried by the valve. When the total upward force on the valve due to the steam pressure exceeds the weight supported by the valve, it lifts and allows steam to escape. The valve, whose face is spherical, is constrained to rise and fall vertically by means of the three steady screws carried by the sleeve, and also by the plate attached to the bottom of the load. The four holes in the sleeve immediately above the valve permit of the ready escape of the steam when the valve lifts.

If d = diameter of valve in inches.
P = pressure in lb. per square inch at which the valve lifts.
W = total dead load attached to the valve.

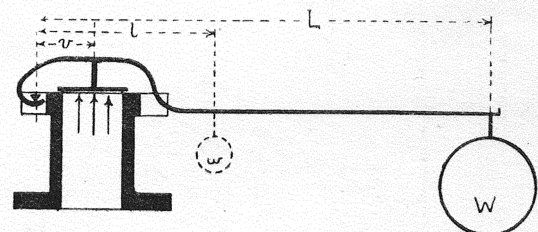
then, $d^2 \frac{\pi}{4} \times P = W$.



LEVER SAFETY VALVE. Figs. 5, 6, 7, 8. The cast-iron body consists of a short length of pipe with a lower circular flange—which is secured by four bolts to a steel seating block—and an upper one of elliptical form. The brass valve is conical, and is held against the brass seating by a spindle fixed into the lever.

It is required to hold down the valve until the pressure per square inch in the boiler, and therefore on the under side of the valve, reaches a definite amount. By means of the lever arrangement shown in the drawing a sufficiently high downward pressure is produced on the valve by means of a comparatively small weight placed at the end of the lever. The steel fulcrum, against which the curved end of the lever presses, is fixed across a slot in the elliptical flange. For stability the lever is curved, so that the centre of gravity of its own and the movable weight may be below the fulcrum.

In order that the valve may "blow off steam" at a given pressure, say "P" lb. per square inch, the valve of the balance weight W for a given position on the lever, may be determined as follows :—



if d = diameter of valve in inches,
and P = steam pressure in lb. per square inch,
then, total upward pressure on the valve $= P \times \frac{d^2\pi}{4}$ lb.

The upward pressure must be balanced by the downward pressure produced by the lever and weight.

The weight of the lever, w lb., will assist W in resisting the steam pressure, and if the centre of gravity of the lever be l inches from the fulcrum, then, applying the principle of moments,

$$P \times \frac{d^2\pi}{4} \times v = WL + wl.$$

If all the above values except W be known, then W may be determined by the above formula.

EXERCISES.

1.—**Dead-weight Safety Valve.** Draw the given sectional elevation and plan, and add an outside elevation and plan. *Scale* 3″ = 1 *foot*.

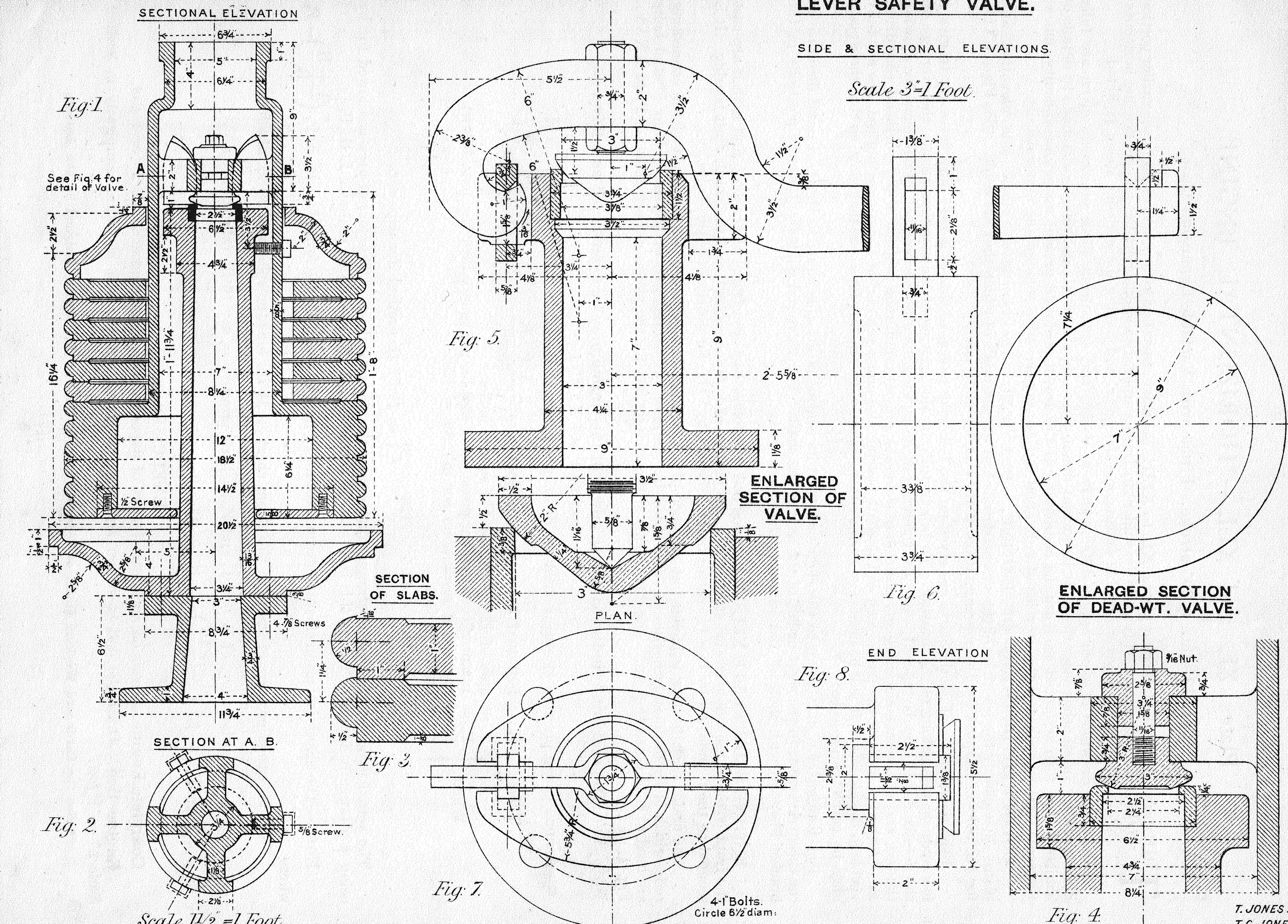
2.—**Lever Safety Valve.** Draw the side and the end elevation and plan completely. *Scale* 4″ = 1 *foot*.

* *The drawings on the accompanying plate represent safety valves made by Messrs. Yates and Thom, Blackburn.*

DEAD-WEIGHT SAFETY VALVE.

LEVER SAFETY VALVE.

SIDE & SECTIONAL ELEVATIONS.



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Plates XL. and XLI. — *HORIZONTAL ENGINE.

Cyl. 12″ dia. × 20″ stroke. Fitted with Variable Expansion Gear.

THE drawings on the following plates, XL. to XLIII. inclusive, together with others given in the earlier portion of the book, are such as will enable the student to draw completely a single cylinder, non-condensing horizontal engine, fitted with Meyer's variable expansion valve gear, and having the cylinder steam jacketed.

Only the drawings of parts which differ widely in design and scale from those previously given are fully dimensioned. In all other cases the most important dimensions only are given, and reference is made to details of similar design. To advanced students the completing of the drawings should present no serious difficulty, but at the same time will allow of the exercise of individual skill in proportioning some of the minor details.

COLOURED DRAWING OF ENGINE. PLATE XL.—The two given views—side elevation and plan—show clearly the positions of the several details and the materials of which they are made. (See Plate LXIX. for the colours adopted for the representation of the various materials.)

Brief descriptions of all the parts are given on the following pages, so that it is here only necessary to give a few hints on the making of the drawing. To benefit fully by the drawing of the complete engine, the exercises on the details should first be worked, and then reference made to them as often as possible. Commence by drawing the bed, and afterwards, assuming any position of the crank pin, determine the position of the crosshead, eccentrics, &c. Although some of the parts, such as the shaft bearing and crank pin end of the connecting rod, are not fully visible in the elevation, it will be necessary to draw them completely—in thin lines—in that view, so that their plans may be correctly projected.

The distance of a slide valve to the right or left of its central position is equal to the horizontal distance of the eccentric centre to the right or left of the axis of the shaft. The valve rod guides will occupy positions in the bracket identical with those of the valves in the steam chest.

ENGINE BED. PLATE XLI. –The engine bed is a hollow cast-iron frame of $\frac{3}{4}''$ metal, and with it are cast the front cover of the cylinder with stuffing box, the bottom slide for crosshead, and the crank shaft bearing. It is secured to the foundation by five $1\frac{1}{8}''$ cotter bolts. (See Plate VIII. for details of cotter bolts.)

The crank shaft bearing is inclined at an angle of 45° so that the maximum pull or thrust of the connecting rod—which is approximately horizontal—is resisted by the most substantial part of the step. Four $\frac{7}{8}''$ stud bolts are used to hold down the cap. The dimensions of the stuffing box for the piston rod are given on Plate XXXV., Fig. 1.

The maximum indicated horse power of the engine may be determined as follows :—

[see text to Plate XXXVI.]

Assume gauge pressure at steam chest = 70 lb. per square inch—

Then H = indicated horse power of engine $= \dfrac{2\,A\,P_m\,R\,S}{33000}$

Where
- A = area of 12″ piston = 113 square inches.
- P_m = **mean** effective steam pressure = 43·5 lb. per square inch (see note below).
- R = number of revolutions per minute = 120.
- S = stroke in feet $= \frac{20}{12}$.

$$H = \frac{2 \times 113 \times 43{\cdot}5 \times 120 \times 20}{33000 \times 12} = 59{\cdot}57 \text{ theoretical horse power.}$$

The actual indicator diagram is always smaller than the theoretical one. This loss of area is allowed for by using a **diagram factor.** Taking this factor as ·85 then 59·57 × ·85 = 50·6 = probable I. H. P.

NOTE.—The theoretical mean effective steam pressure P_m may be found by the formula. $P_m = P_1\left(\dfrac{1+H}{R}\right) - P_b$

where

P_1 = gauge pressure of steam in the steam chest + 15 lb. for atmosphere = 85 lb. absolute.
R = ratio of expansion = 3 ; steam being cut off at $\frac{1}{3}$ stroke.
H = hyperbolic log. of R = 1·1, and P_b = back pressure = say 16 lb. (absolute).

Then $P_m = 85\left(\dfrac{1 + 1{\cdot}1}{3}\right) - 16 = 43{\cdot}5$ lb.

EXERCISES.

Complete Engine.—Draw on an imperial sheet of paper the two given views. *Scale 2″ = 1 foot.*

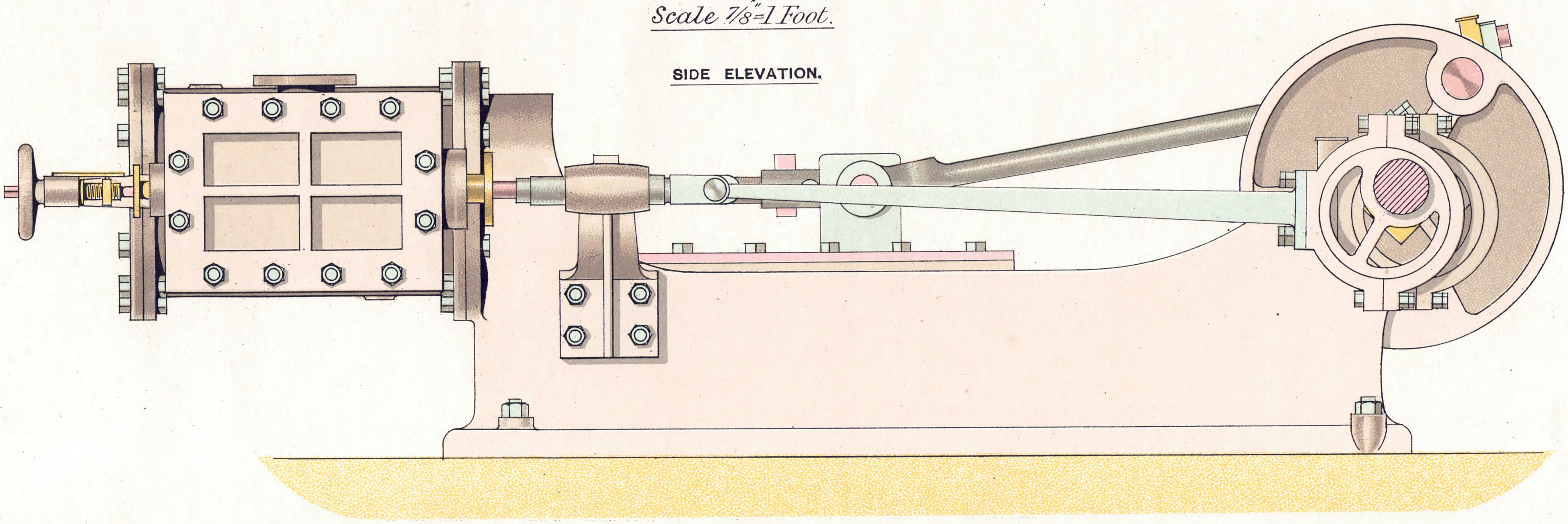
Engine Bed.—Draw the end elevation and plan as given, but the side elevation completely in section *Scale 2″ = 1 foot.*

* See the Cardboard Working Model of Complete Engine, No. 2, "X," Series, designed by the Authors.

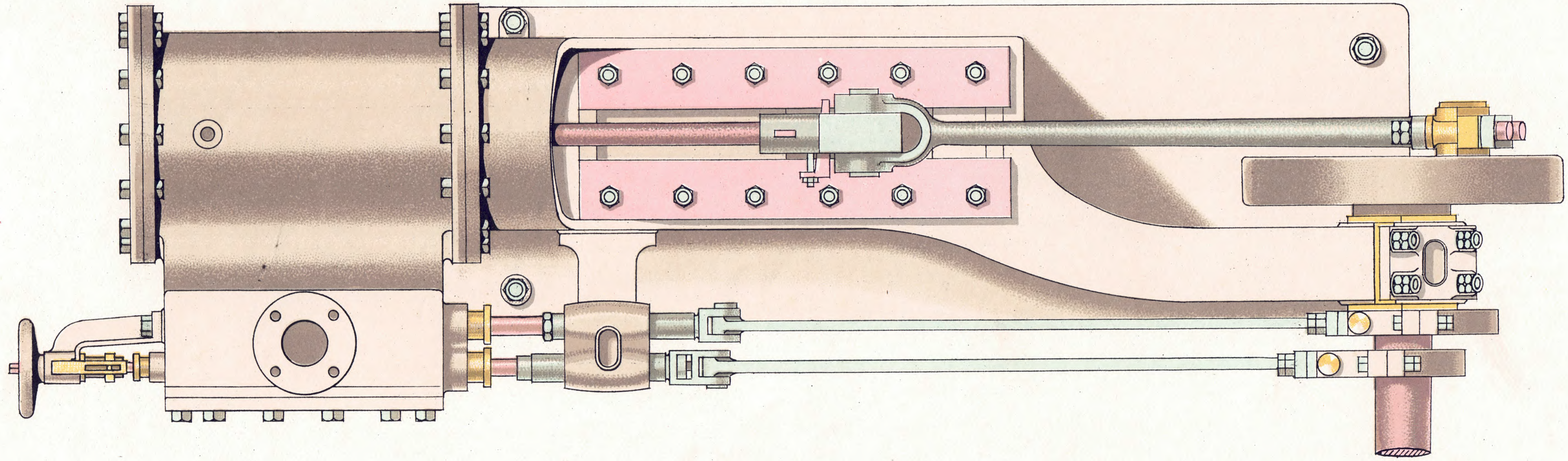
HORIZONTAL ENGINE WITH VARIABLE EXPANSION GEAR.

Scale 7/8" = 1 Foot.

SIDE ELEVATION.



PLAN.



NOTE.—For dimensioned details, see Plates Nos. 25, 26, 35, 37, 41, 42 and 43.

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DETAILS OF 12" × 20" HORIZONTAL ENGINE.

ENGINE BED.

Scale 1"=1 Foot.

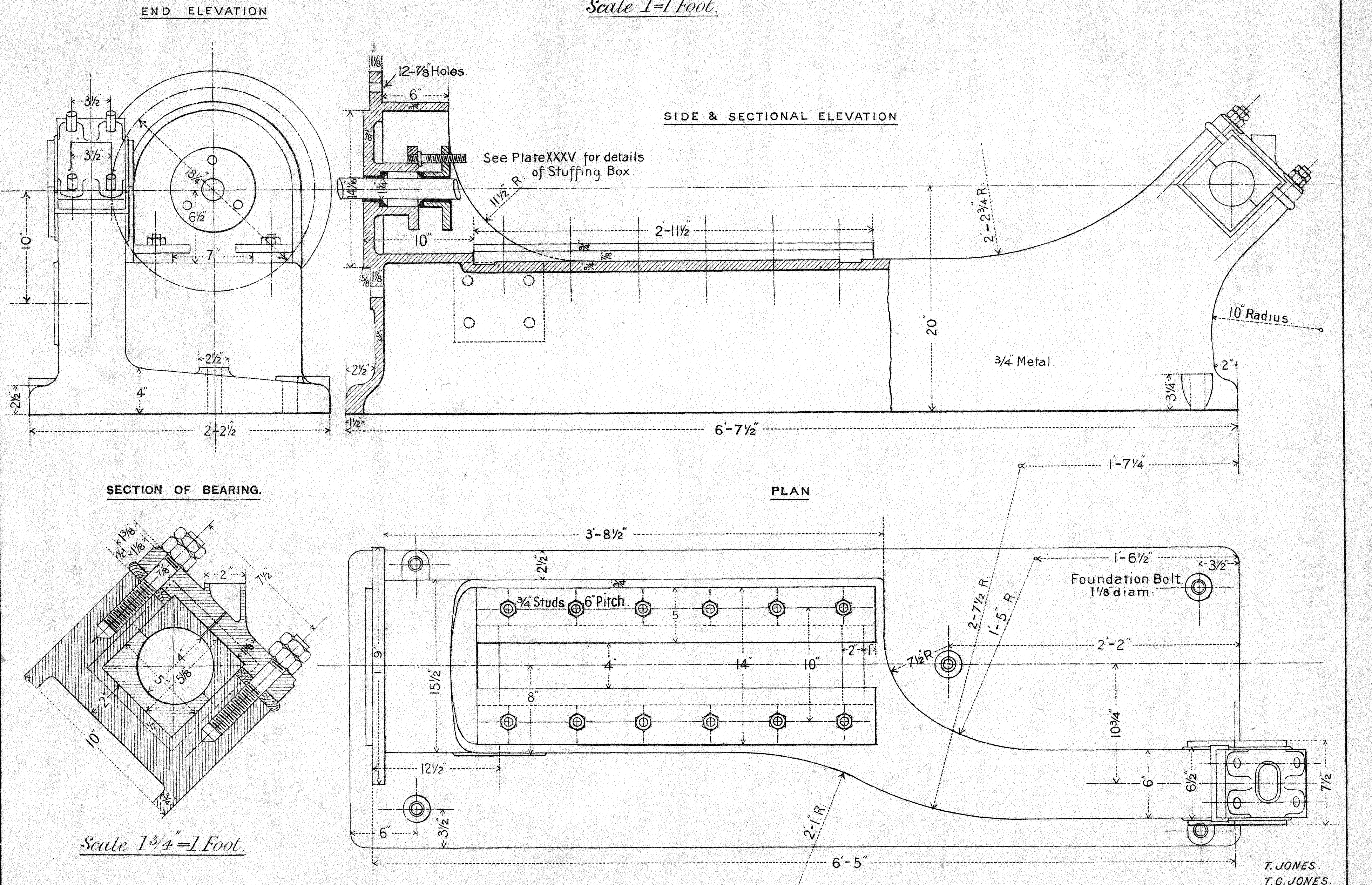

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Plate XLII.—DETAILS OF HORIZONTAL ENGINE.

STEAM CYLINDER. **Plate XLII.**—The cylinder, which overhangs the bed, is secured to its end by bolts and studs. The liner is made of close grained cast-iron, and is slightly tapered on the outside; it is driven into position in the cylinder from the back end.

The space between the liner and the cylinder body forms the steam jacket, which is supplied with "live" steam. The condensed steam is drained from the jacket by means of a steam trap. The function of the jacket is the prevention of excessive condensation of steam in the cylinder during admission and early expansion.

In this engine the steam chest is nearly as long as the cylinder. This is caused by using Meyer's variable cut-off valves to effect the steam distribution. To shorten the external length of the chest the vertical flanges are cast on the inside. The steam is supplied through a $3\frac{1}{2}''$ pipe and exhausted through a $4''$ pipe.

SLIDE VALVES AND RODS. Plate XLII. For the **design** of the main and expansion valves see Plate XXXVII.

The **main** valve is $3\frac{1}{2}''$ deep, and the two steam passages—each $1\frac{1}{4}''$ wide—are at **right angles** to the face of the valve. Its dimensions, as affecting the steam distribution, are as follow:—For the forward stroke of the piston: steam lap $1\frac{1}{4}''$, exhaust lap $\frac{1}{4}''$, lead $\frac{1}{8}''$; for the return stroke: steam lap, $1\frac{3}{16}''$, exhaust lap $\frac{1}{4}''$, lead $\frac{3}{16}''$.

Each expansion plate is $3\frac{1}{2}''$ wide across the face. By regulating the distance between their outer edges, the steam may be cut off at any part of the stroke earlier than that due to the main valve alone, which effects the **cut off at 0.75 of forward stroke and 0.7 of return stroke.**

The method of changing the point of cut off during the running of the engine is explained in the text to Plate XXXVII.

The valve rods outside the steam chest are enlarged by their connection with wrought-iron cylindrical guides, which have forked ends for attachment to the eccentric rods. These guides are supported by a cast-iron bracket fixed to the side of the engine bed.

ECCENTRICS AND RODS. See Plate XXVI., Fig. 2, for design of an eccentric. For this engine the main dimensions are as follow:—Width of eccentric sheave and strap, $2''$; boss, $2\frac{1}{4}''$ through; eccentricity, $2\frac{1}{2}''$.

The T end of each eccentric rod is $6\frac{1}{2}''$ long $\times$ $2''$ wide; each is fixed to an eccentric strap by two $\frac{7}{8}''$ studs $4\frac{1}{4}''$ apart. The section of the rod at the strap end is $2\frac{1}{2}'' \times 1''$, and that at the valve rod end $1\frac{1}{4}'' \times 1''$.

The effective lengths of the eccentric rods for the main and expansion valves, measured from the centres of the eccentric blocks, are $4' - 8''$ and $4' - 6\frac{1}{4}''$ respectively: and assuming $6\frac{1}{2}''$ to be the distance between the centre of the eccentric and the face on the strap to which the T end of the rod is attached, then the respective real lengths of the rods are $4' - 1\frac{1}{2}''$ and $3' - 11\frac{3}{4}''$.

The eccentric for the main valve has an angular advance of 34°, and that for the expansion valve is placed diametrically opposite to the crank.

DISC CRANK. For design see Plate XXV., Fig. 3. The disc is $4''$ across the rim, $2' - 2\frac{1}{2}''$ outside diameter, and $5''$ through the boss.

The crank pin is of steel $3\frac{1}{4}''$ diameter $\times$ $4''$ long between the collars. The disc is shrunk to its end, and the pin further secured by means of a key.

EXERCISES.

1.—**Steam Cylinder.** Draw the two given sectional views, and add a side and an end elevation. Omit the valves, but show the valve spindles. *Scale* $3'' = 1$ *foot.*

2.—**Main and Expansion Valves.** Refer to the notes given above for the more important dimensions of the valves, and to Plate XXXVII. for their design, and then draw the back and end elevations and sectional plan. *Scale* $\frac{1}{2}$ *full size.*

3.—**Expansion Valve Spindle Bracket.** Draw a side elevation (with hand wheel in position), end elevation and sectional plan. *Scale* $\frac{3}{4}$ *full size.*

4.—**Valve Rod Guides and Bracket.** Draw the front and end elevations and plan, showing the guides in position and portions of the valve and eccentric rods. *Scale* $4'' = 1$ *foot.*

5.—**Eccentrics and Rods.** Draw the side elevation and plan of the two eccentrics and rods in correct relative positions, assuming the crank to be on the "dead centre." *Scale* $3'' = 1$ *foot.*

6.—**Disc Crank.** Draw sectional and back elevations. *Scale* $\frac{1}{3}$ *full size.*

DETAILS OF 12″ × 20″ HORIZONTAL ENGINE.

CONNECTING ROD AND CROSSHEAD.

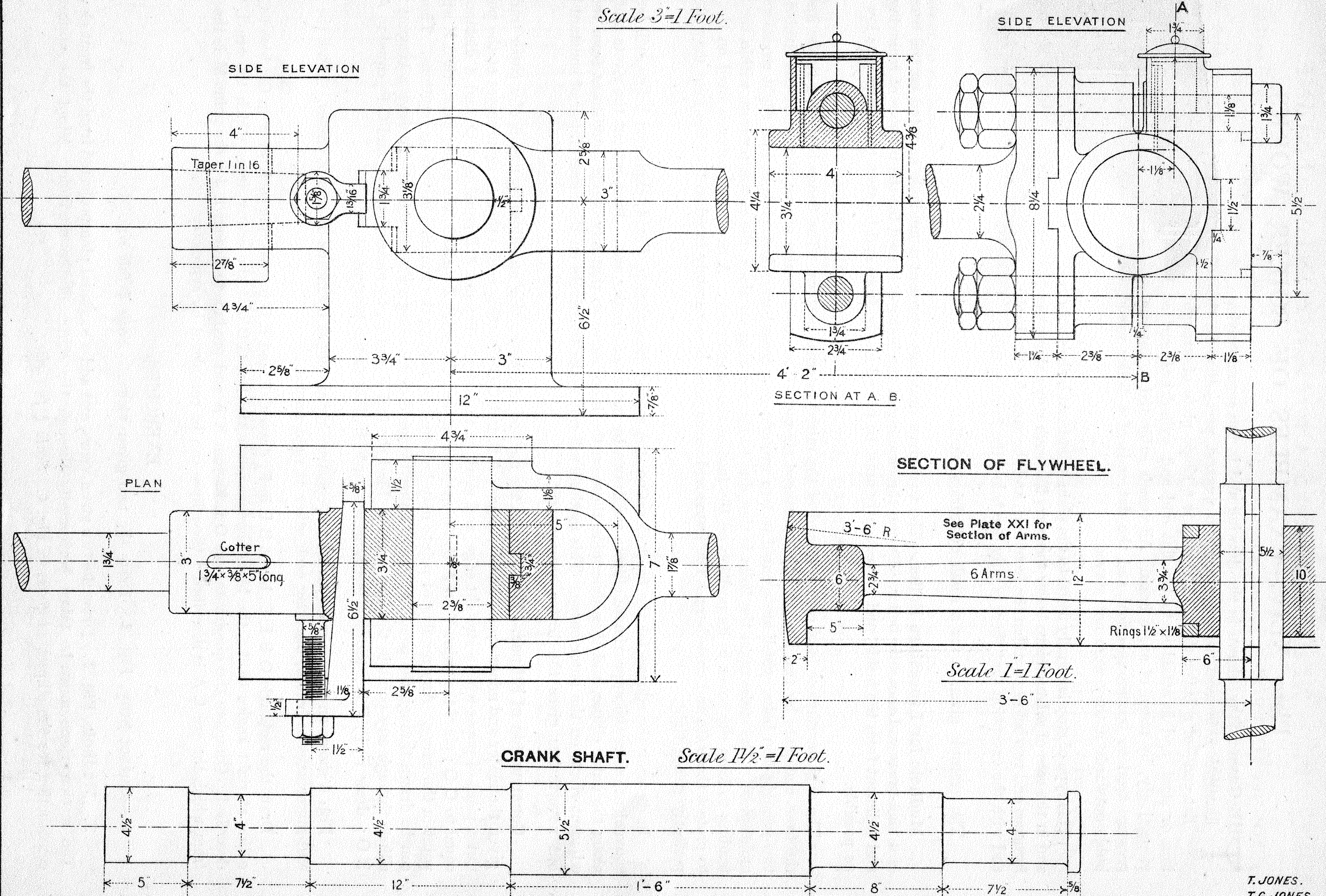

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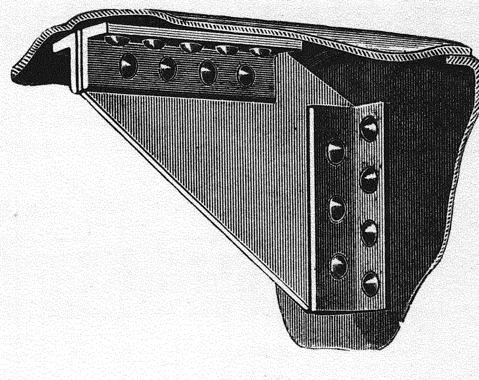
Plate XLIV.—EXAMPLES OF BOILER WORKS.



THE **Gusset Stay,** shown in the drawing and in the accompanying illustration, is used to strengthen the weakest part—that is, the flat end—of a boiler shell. The stay is a single plate of wrought iron or steel, and is attached to the flat end and cylindrical surface respectively, by mean of angle irons. All vessels subjected to internal pressure tend to become spherical, and the flat ends of a boiler would bulge outwards if they were not stayed.

This type of stay is usually fitted to the plain cylindrical boilers only, and in the case of a Lancashire boiler there are five such stays above the furnaces and two below to stiffen the front end, and the same for the back end with the exception that there is only one below the furnace tubes.

The example given is for a small cylindrical shell.

Mudhole for Lancashire Boiler.—The two given drawings represent a mudhole as fitted on the front end of a high-pressure Lancashire boiler. During the cleaning of the boiler it is used for the purpose of removing scale and sediment from the inside, and consequently must be placed on the front end between the furnaces as low down as possible.

The oval mouthpiece is of cast steel and of the raised internal pattern. The flange is riveted to the inside of the front end plate by two lines of rivets arranged zig-zag. The cover is also of cast steel, and has riveted to it two wrought-iron bolts, which are secured to two wrought-iron cross bars, whose ends fit into slight recesses in the mouthpiece. The stress upon the bolts is that due to the screwing up alone, the steam pressure within the boiler tending to keep the cover in its position on the face of the mouthpiece. A neat steel cover is fixed on the front to protect the bolts and cross bars. The frame and cover must be accurately faced so that the joint may be made perfectly steam tight.

In drawing the ellipses for the front elevation, the second construction, Fig. 4, explained below, will be more convenient, since the approximate ellipses are drawn by means of circular arcs. First set out the rivets uniformly along the outer ellipse, and then determine those on the inner curve so that any one is equidistant from the two adjacent rivets on the outer curve. This method will not make the rivets on the inner ellipse of constant pitch, but nevertheless gives the better arrangement.

Construction of Ellipse.—Fig. 3 shows the method of constructing an ellipse by means of a paper trammel; given the major and minor axes. Draw the axes A B and C D bisecting each other at right angles at the point O. Cut a piece of paper with a straight edge, and mark on this edge a distance X Z equal to A O, and a distance X Y equal to C O; place the point Y on A B, the major axis, and the point Z on C D, the minor axis; then X will be a point in the curve. Any number of points may be found by placing the trammel in various positions. The curve can be drawn through the points so found, either freehand or by using a French curve. To draw a tangent to the ellipse through any point in the curve as P. Find the foci F and F_1 by measuring C F and C F_1 equal to A O or B O; draw F_1 P, F P; the line bisecting the angle F_1 P Q is the tangent at P.

Fig. 4 shows an approximate method of constructing an ellipse by means of arcs of circles. Draw the major and minor axes as before, bisecting each other at right angles.

Draw the rectangle O B E C, bisect B E in F, draw C F and E D intersecting each other in G; bisect C G by a line at right angles to it; the bisecting line meets the line C D in J, the centre, from which the arc L C G is described. Complete the quadrantal arc C L K, join K A and produce to L, join L J; the point M is the centre from which the arc L A is described; the ellipse can be completed by symmetry.

EXERCISES.

1.—Gusset Stay, Fig. 1. Draw the two given views. *Scale $\frac{3}{8}$ full size.*

2.—Mudhole, Fig. 2. Draw the two given views, and add a sectional plan projected from the front elevation. The section plane must be taken horizontally through the centre of the mudhole. Adopt the approximate construction for the ellipses as given in Fig. 4. *Scale $\frac{3}{8}$ full size.*

GUSSET STAY FOR BOILER END

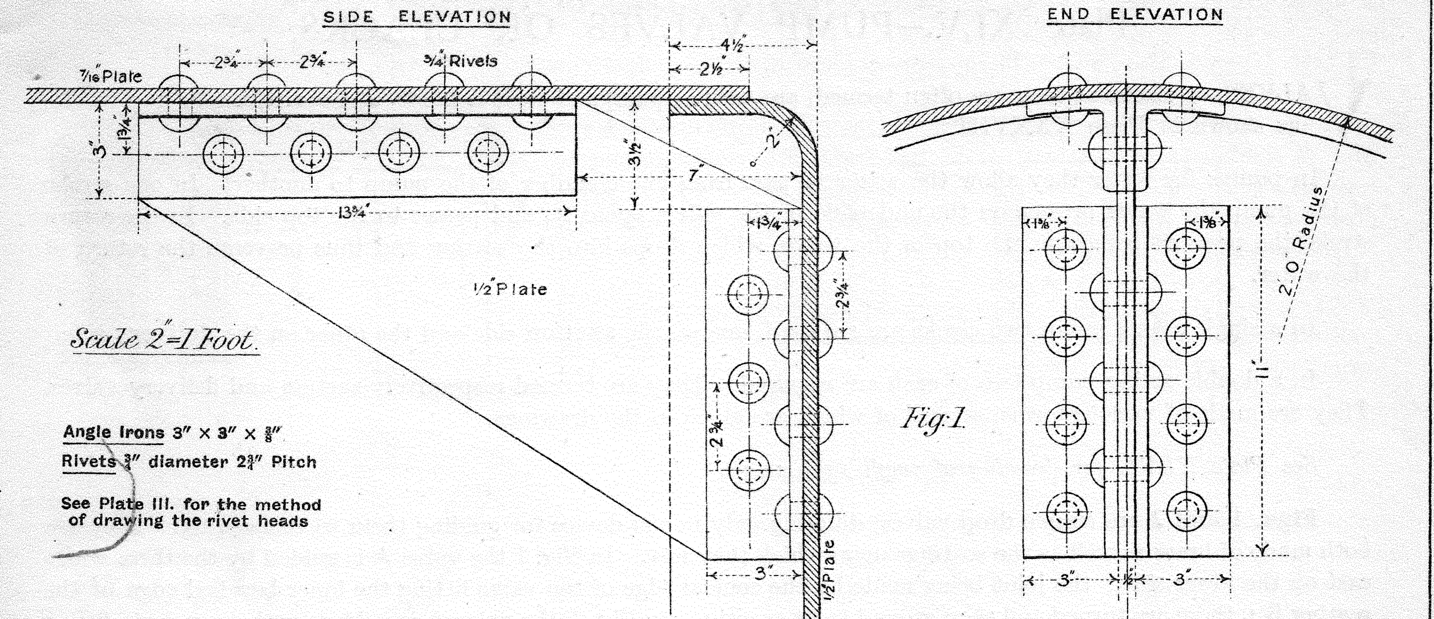


Fig: 1.

MUDHOLE FOR HIGH-PRESSURE LANCASHIRE BOILER

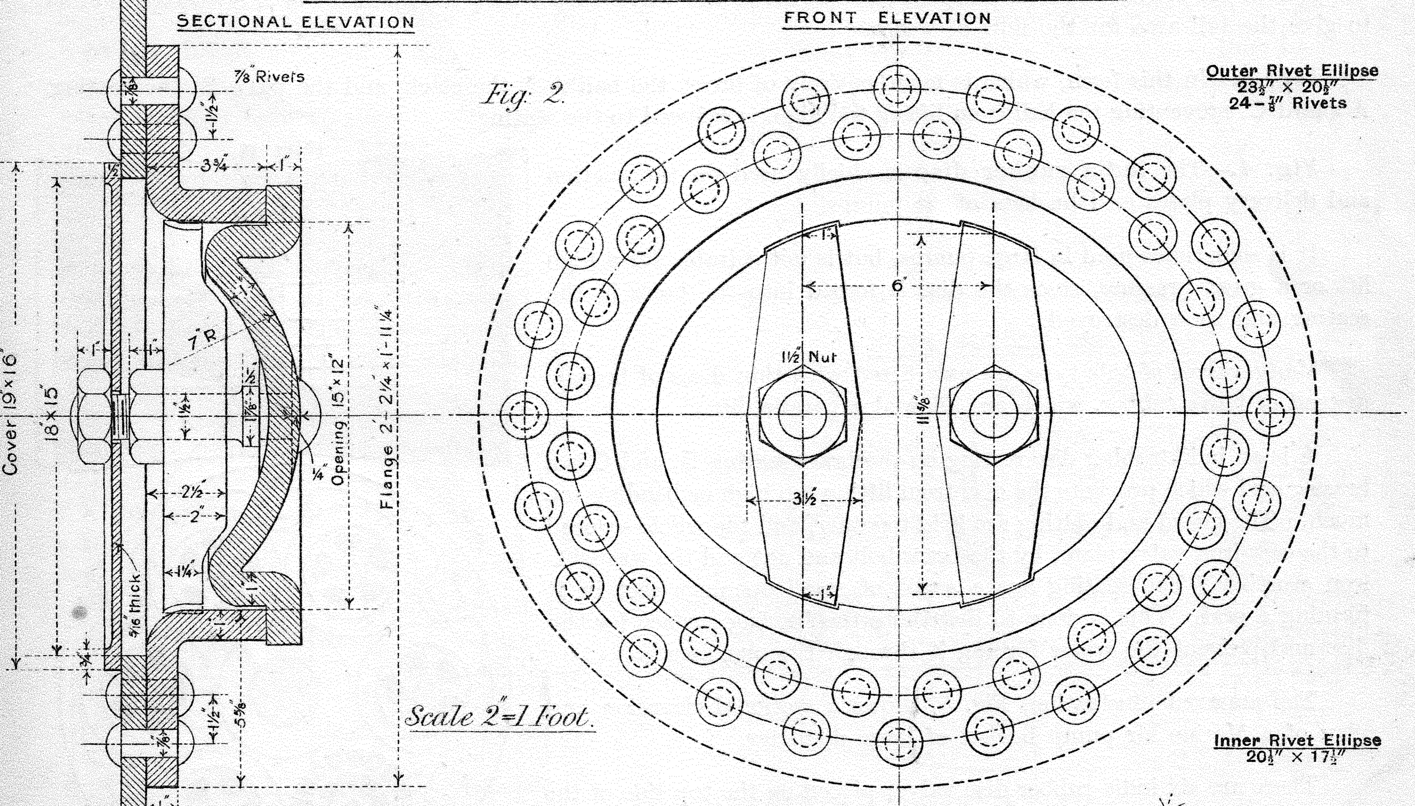


Fig: 2.

CONSTRUCTION OF ELLIPSE

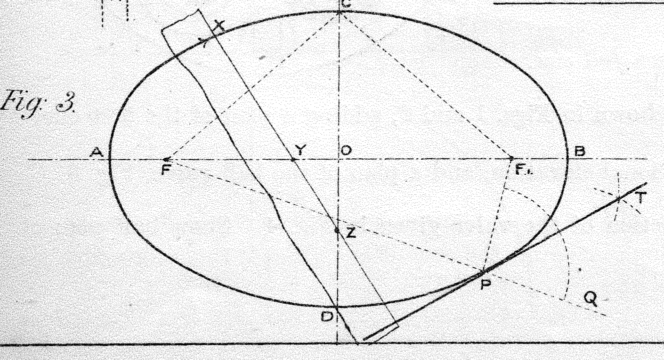


Fig: 3.

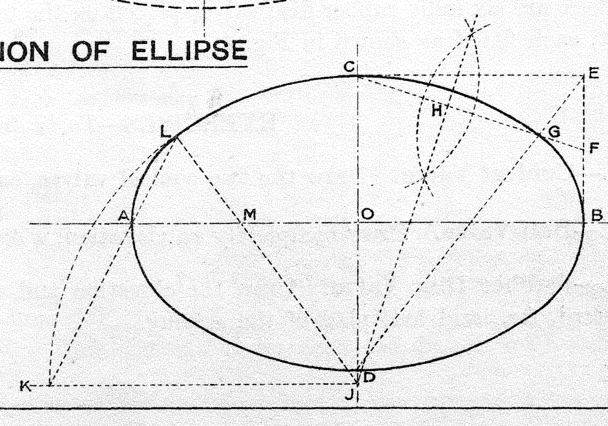


Fig: 4.

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Plate XLV.—PUMP VALVES OR CLACKS.

VALVES, or clacks as they are often termed, are used in pumps of various kinds, and sometimes in stop valves, as shown in Plate XXXVIII.

In pumps for water they allow the water to pass from one chamber of the pump to another. In one stroke of the pump the water presses on the underside of the valve, opens it, and passes to the top side. In the return stroke the pressure comes on the top of the valve, which drops into the seating and thus prevents the return of the water.

In a single acting pump two clacks are required, one on the **suction** side and the other on the **delivery** side.

In a double acting pump two of each are required. They are termed respectively suction and delivery valves. They are made of various forms, several of which are given in the drawings.

See Plate XLVI. for drawing of small ram pump.

Figs. 1 and 2 are simple drop valves, differing only in the device for guiding them to the seating. They are both made of brass as well as the seatings upon which they rest. In Fig. 1 the valve A is guided by the three wings cast on the lower side ; the joint being made by the conical edge of the valve fitting the inner bevelled edge of the seating B ; these are turned and then ground to fit exactly. In Fig. 2, the valve A is guided by the round pin fitting the boss in the centre of the seating. The *clear* lift of the valve should be $\frac{1}{4}$ the diameter of the passage in order to give the full area for the flow of water.

Fig. 3.—In this form, which is made entirely of brass, the ball A is the clack, and the piece B, the seating. A guard C, preventing the ball from lifting too high, is screwed to the seating.

Fig 5.

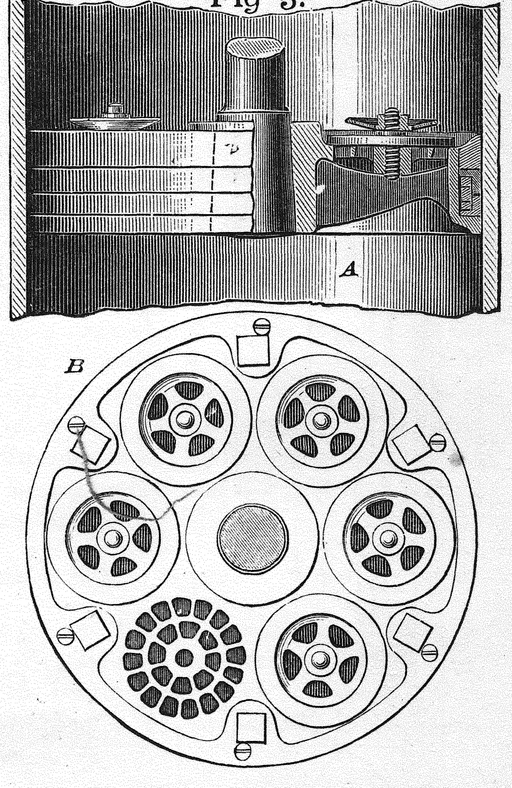


Fig. 4.—The **india rubber disc** is chiefly used on the suction and delivery plates and buckets of air pumps.

It is sometimes used in large pumps, but is not suitable for a high lift or a great pressure, since the disc is forced into the holes in the seating and soon destroyed.

Many valves of this type are now fitted with thin discs of bronze, instead of india rubber, which are allowed a parallel lift.

A is an india rubber disc resting on the brass seating B, and C is a brass guard which prevents the disc from lifting too high or bending too much. The guard and seating are held together, and the whole secured to the cast-iron valve plate, by the brass bolt and nut and the wrought-iron crossbar. The seating has a series of small openings cast in it forming a grid. This is done to distribute the bearing surface for the disc and prevent the water forcing it through the openings.

The annexed illustration, Fig. 5, shows a sectional elevation and plan of a Marine air pump bucket of large diameter.

There are six india rubber disc valves placed on the top side of the bucket, each fitted as shown in Fig. 4.

EXERCISES—*To be drawn full size.*

1.—**Conical Valve.** Draw the two conical valves, as shown in Figs. 1 and 2, adding a plan of the first one.

2.—**Ball Valve.** Draw completely an elevation, a sectional elevation, and a plan of the ball valve, Fig. 3.

3.—**Rubber Disc Valve.** Draw the elevation and section of the valve given in Fig. 4. Show half plan of the guard, &c., and half plan of the seating.

SIMPLE VALVES FOR PUMPS.

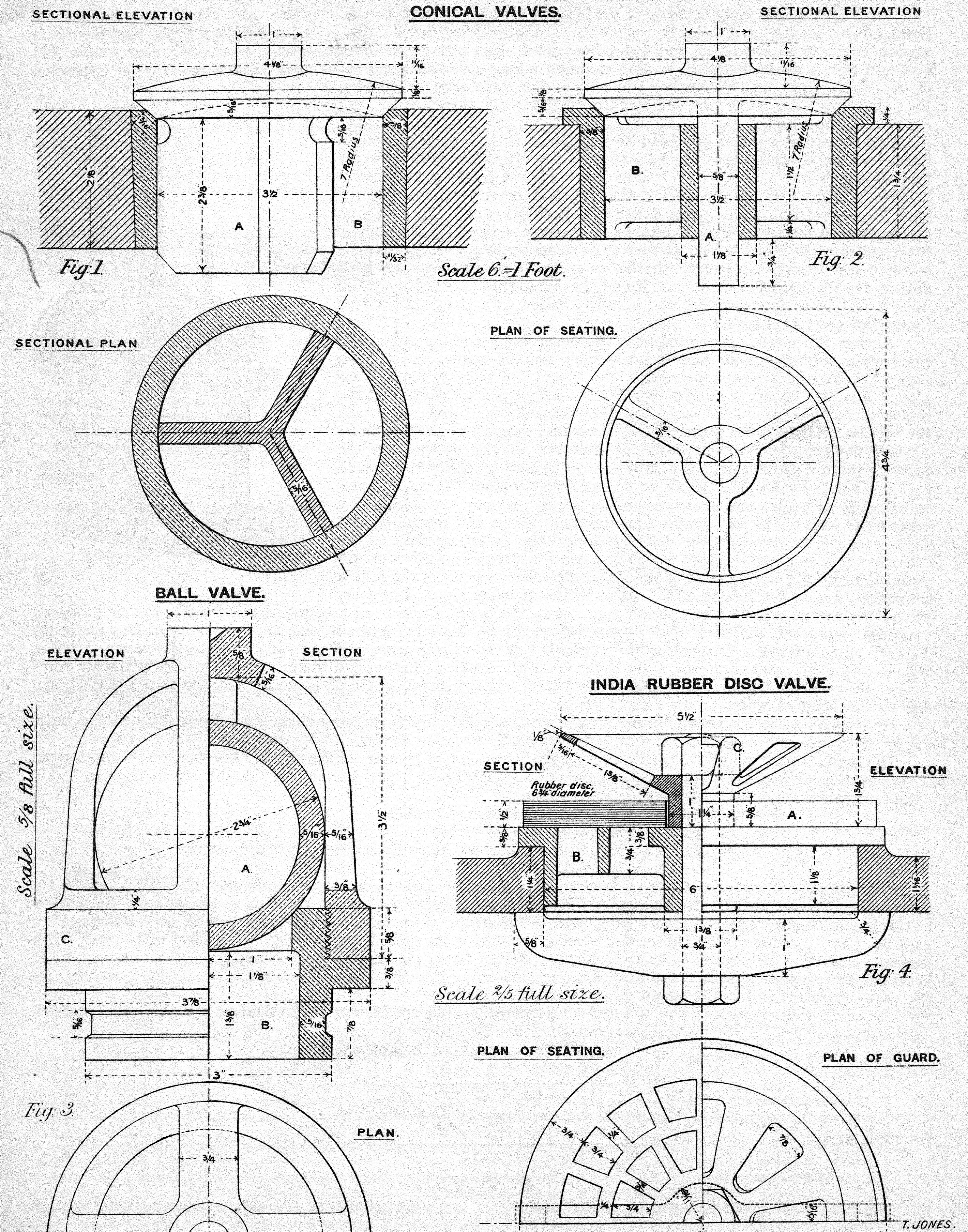
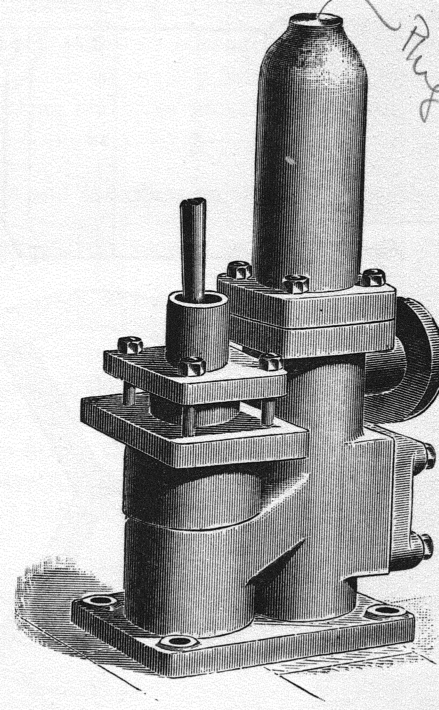


Plate XLVI.—RAM PUMP.

THE drawings on the accompanying plate represent one of three single-acting force pumps which are actuated by means of a three-throw crank shaft, driven by gearing from an electrical motor or a small steam engine.

The cast-iron body consists of the barrel, in which the ram works, and the valve chamber containing two brass valves—suction and delivery respectively. The packing for the ram is of the ordinary form, consisting of a stuffing box with square flange and a cast-iron gland—also with square flange—held in position by four studs. The cast-iron ram is of the trunk type, thus enabling a long connecting rod to be used without making the centre-line of the crank shaft inconveniently high, and at the same time minimising the obliquity of the connecting rod and the resulting side thrust of the ram against the packing.



Plug!

The air vessel, which is bolted to the top flange of the valve chamber, is formed with a central pin at the inlet to limit the lift of the delivery valve immediately below. Access to the suction valve is provided by the vertical hand-hole and cover on the side of the valve chamber, and the internal horizontal projection on the cover limits the lift of the valve. To obtain an opening round the valve equal in area to the passage covered by it the lift of the valve must be equal to one quarter of its diameter, but generally, the lift is made less than this to diminish the quantity of water that runs back during the closing of the valve. From the arrangement of the suction inlet it will be understood that the pump is bolted to a base-plate, which forms the suction chamber.

Action of Pump. Assuming that the pump is in working order, i.e., the barrel, valve chamber and delivery pipe contain water, and the air vessel, air at a pressure corresponding to the " head " of water in the delivery pipe; then, on the **up** or **suction stroke** the delivery valve closes and the atmospheric pressure, on the surface of the water supply, forces water past the suction valve into the barrel to fill the volume vacated by the ram in its upward motion. During the **down** or **delivery stroke** of the ram the suction valve remains closed, and the water displaced by the ram is forced past the delivery valve into the air vessel and delivery pipe. Since the ram is actuated by a crank and connecting rod its velocity is very variable—being zero at the ends of the stroke and a maximum at about half stroke—and if there were no air vessel on the delivery side of the pump, as close to the delivery valve as possible, there would be excessive stresses on the ram and connections during the first half of the stroke when the velocity of the ram is increasing, due to the inertia of the water in the delivery pipe. However, when the pressure on the ram exceeds that due to the head of water, on account of this inertia, the air in the air vessel is compressed, and some of the water delivered past the valve enters it, and so the velocity of flow along the delivery pipe during the first half of the stroke is less than that corresponding to the velocity of the ram. Later, the velocity of the ram decreases, and the inertia of the water in motion and the increased pressure in the air vessel enable the remainder of the stroke to be performed without shock, and with a pressure on the ram less than that due to the head of water.

By its action the air vessel produces an approximately uniform delivery since a small quantity of the water displaced by the ram is discharged during the succeeding suction stroke.

The larger the air vessel, the smaller will be the fluctuation of pressure of the air, and the steadier the discharge.

Quantity of Water Delivered.—The theoretical quantity of water delivered per double stroke is equal to the volume displaced by the ram on the down stroke.

If A = area of ram in square inches,
and S = stroke of ram in inches,
and q = theoretical discharge in cubic inches per double stroke,
then, $q = A \times S$.

The quantity of water actually delivered is less than the theoretical quantity because of the leakage at the valves when closed, and the gradual and not instantaneous closing of them at the ends of the stroke. In addition to the loss of efficiency due to the running back of the water, there may, with plunger pumps, be a leakage of air past the gland packing during the suction stroke, preventing the cylinder space from being filled with water. The passage connecting the barrel and valve chamber should be so placed that an accumulation of air is impossible. With the arrangement shown in the drawing, any air leaking into the barrel passes along the inclined passage into the valve chamber, and is discharged on the next down stroke.

For small pumps, such as the one under consideration, the **coefficient of discharge** may be taken at **0.85**, so that if—

N = number of double strokes per minute,
Q = actual discharge in cubic feet per minute,

$$\text{then, } Q = \frac{0{\cdot}85 \times N \times A \times S}{12 \times 12 \times 12} \text{ cubic feet.}$$

For the given pump, $S = 3''$, area of ram diameter $2\frac{1}{4}'' = 4$ square inches, and assuming 100 double strokes per minute,

$$Q = \frac{{\cdot}85 \times 100 \times 4 \times 3}{12 \times 12 \times 12} = 0{\cdot}59 \text{ cubic feet} = 3{\cdot}69 \text{ gallons.}$$

EXERCISE.

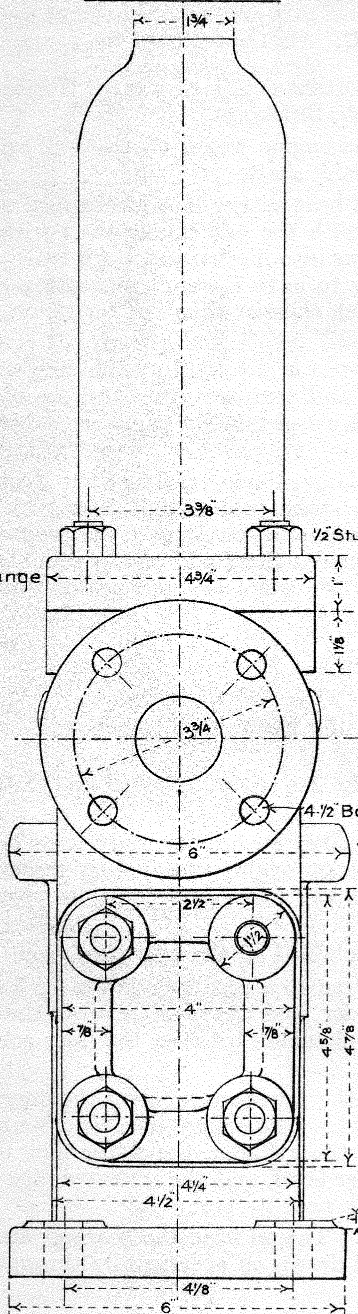
Draw the three given views of the **ram pump** and add a *side* elevation and also a plan projected from it. *Scale ½ full size.*

N.B.—Care must be exercised in placing the views on the paper since the net amount of space required for the elevation and plan is 14″, and the paper measures 15″.

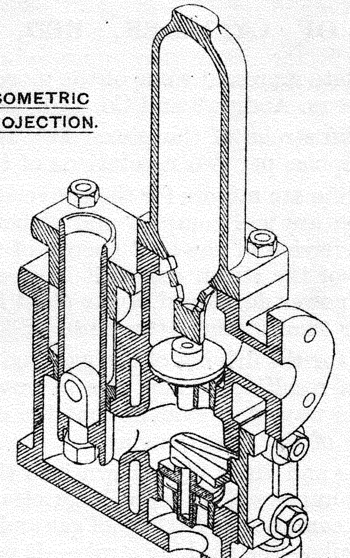
SMALL RAM PUMP (2¼" DIAM., 3" STROKE).

Scale 3½" = 1 Foot.

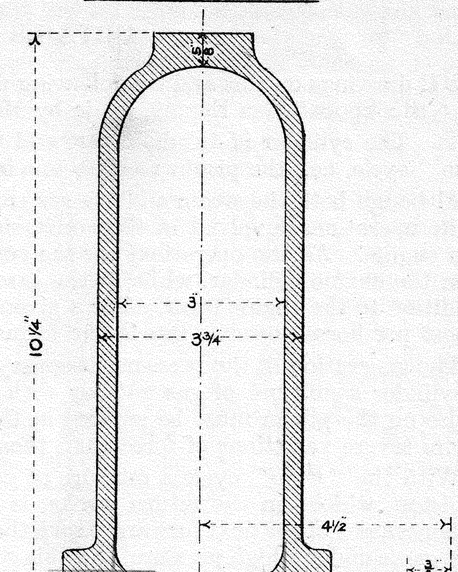
FRONT ELEVATION

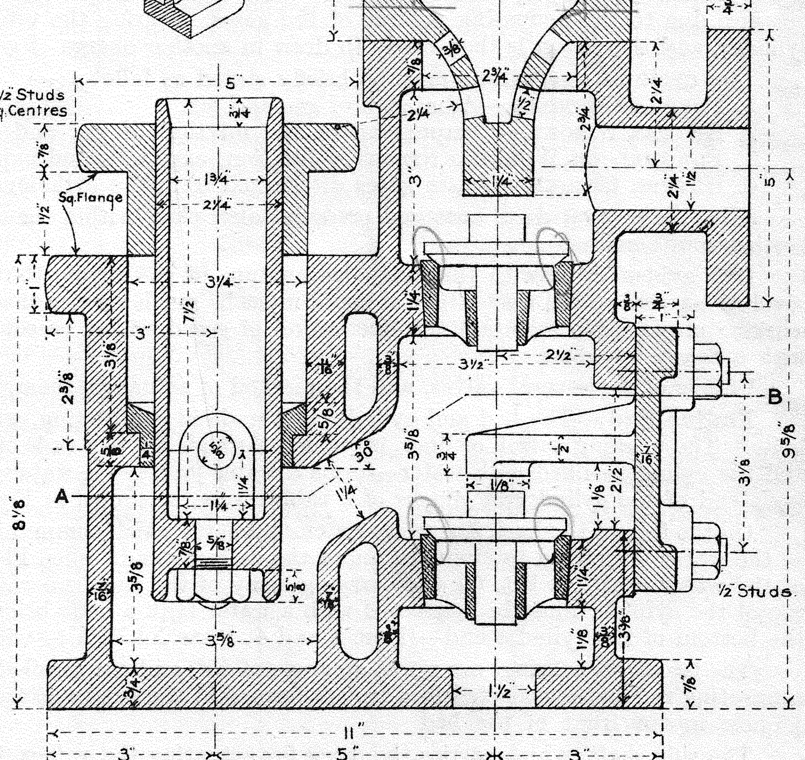


ISOMETRIC PROJECTION.

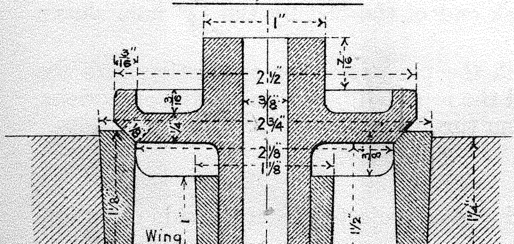


SECTIONAL ELEVATION

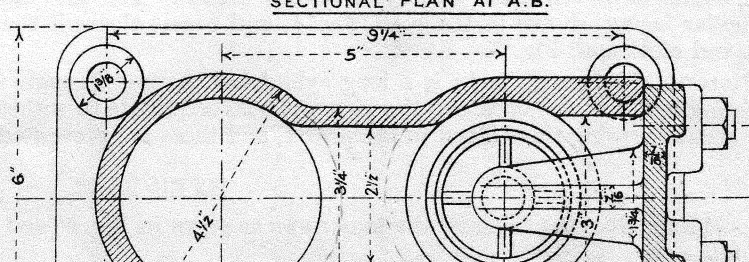




SECTION OF VALVE



SECTIONAL PLAN AT A.B.



Scale ⅝ full size.

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Plate XLVII.—$4\frac{1}{2}''$ × $11''$ GAS ENGINE.

DETAILS OF CYLINDER, BED, ETC.

THE drawings on this and the following plate represent some of the more important details of a small Horizontal "Stockport" Gas Engine, made by Messrs. Andrews and Co. Ltd., Reddish, Stockport.

The cylinder is $4\frac{1}{2}''$ diameter, and the stroke of the piston $11''$, and the engine works on the well-known "Otto" cycle, i.e., the piston receives one impulse per two revolutions of the crank shaft.

Although both the steam and the gas engine are motors for the conversion of heat energy into mechanical work, yet the operations involved in the conversion are less complex and elaborate with the gas engine than with the steam engine. All the operations for the conversion of the heat energy of the gas into mechanical work take place within the engine cylinder, while in the case of the steam engine it is necessary to have a steam generating plant in addition to the engine itself. For a given power developed, a gas plant is much cheaper than one for steam, and the cost per horse-power is less in the former than in the latter case.

The generation of the pressure necessary for the driving of the gas engine piston is effected by exploding within the cylinder a mixture of gas and air with a resulting sudden rise in pressure and temperature; and since each impulse on the piston must be secured in this way, it is evident that the cylinder and moving parts are subjected to more severe variations of force than those of the steam engine.

With the "Otto" cycle a mixture of gas and air is first drawn into the cylinder during the forward stroke of the piston, which, on the return stroke, is compressed into the large clearance space behind the piston. At the commencement of the next forward stroke the compressed mixture of gas and air is ignited, resulting in the production of hot gases under a high pressure, and this stroke—the working stroke—is performed under a continually diminishing pressure due to the increasing volume of the gases. During the return stroke the gases are exhausted from the cylinder, and the piston is then ready to draw in another charge of gas and air.

The cycle of operations may be briefly stated as follows :—

1. FORWARD STROKE.—Admission of gas and air.
2. RETURN STROKE.—Compression of the mixture of gas and air.
3. FORWARD OR WORKING STROKE.—Ignition, explosion and expansion of the gases.
4. RETURN STROKE.—Waste gases discharged from the cylinder.

Since the generation of heat and pressure takes place within the cylinder, the gas engine is called an "internal combustion" engine.

It is evident that with this single-acting type of gas engine the piston receives an impulse only once in two revolutions, and so the fluctuation of speed must be much greater than in the steam engine where every stroke is a working stroke. To maintain the fluctuation of speed within reasonable limits the gas engine must be provided with a heavy fly-wheel.

Details of the several valves, and the method of actuating them, will be explained on the following page.

Engine Bed.—The bed and cylinder are made in one casting, which is bolted to a firm foundation by two $\frac{7}{8}''$ bolts. The working barrel, or liner, is pressed through the two ends of the cylinder, and the space between the liner and the cylinder—filled with water—forms a water jacket. A water-tight joint is made between the liner and the front end of the cylinder by means of a rubber ring, as shown in Fig. 2.

On the face of the back flange are six curved ports which communicate with the water space round the liner, and so the water is enabled to circulate round the cylinder end. (See Plate XLVIII.). To ensure efficient circulation of the water in the jacket, for the carrying away of the heat generated within the cylinder, the water pipe to the top of the cylinder must be connected with a water tank a little below the water level, and the $1''$ water pipe from the bottom of the cylinder end—(Figs. 3 and 4, Plate XLVIII.)—with the tank near its base.

The two shaft bearings are set at an angle of 45°, and are detailed in Fig. 5. The oil from the bearings and the connecting rod end is collected in a trough formed in the centre of the bed, and drawn off occasionally through the $\frac{1}{4}''$ hole in the front of the bed.

The side shaft, which carries the cams for actuating the valves, is driven from the crank shaft through a pair of screw wheels at one-half the speed of the engine. The side shaft is supported by a bearing fixed to the rectangular facing, shown in the plan, Fig. 4, and passes through the back end of the bed by the $1\frac{1}{2}''$ hole shown in the end elevation, Fig. 3.

Piston.—The piston body is a long cylindrical casting through which the pin passes for connection with the connecting rod. At the back end the piston is stiffened inside to withstand the repeated blows due to the explosions. There are four spring rings fitted to the piston, and these are prevented from turning in the grooves by small stops.

EXERCISES.

1.—**Shaft Bearing.** Draw the two views as given in Fig. 5, and add a plan projected from the side elevation. *Scale full size.*

2.—**Piston.** Draw the end elevation, longitudinal sectional elevation and the plan. *Scale full size.*

3.—**Bed and Cylinder.** Draw the three given views, and complete the bearings. *Scale* $3'' = 1$ *foot.*

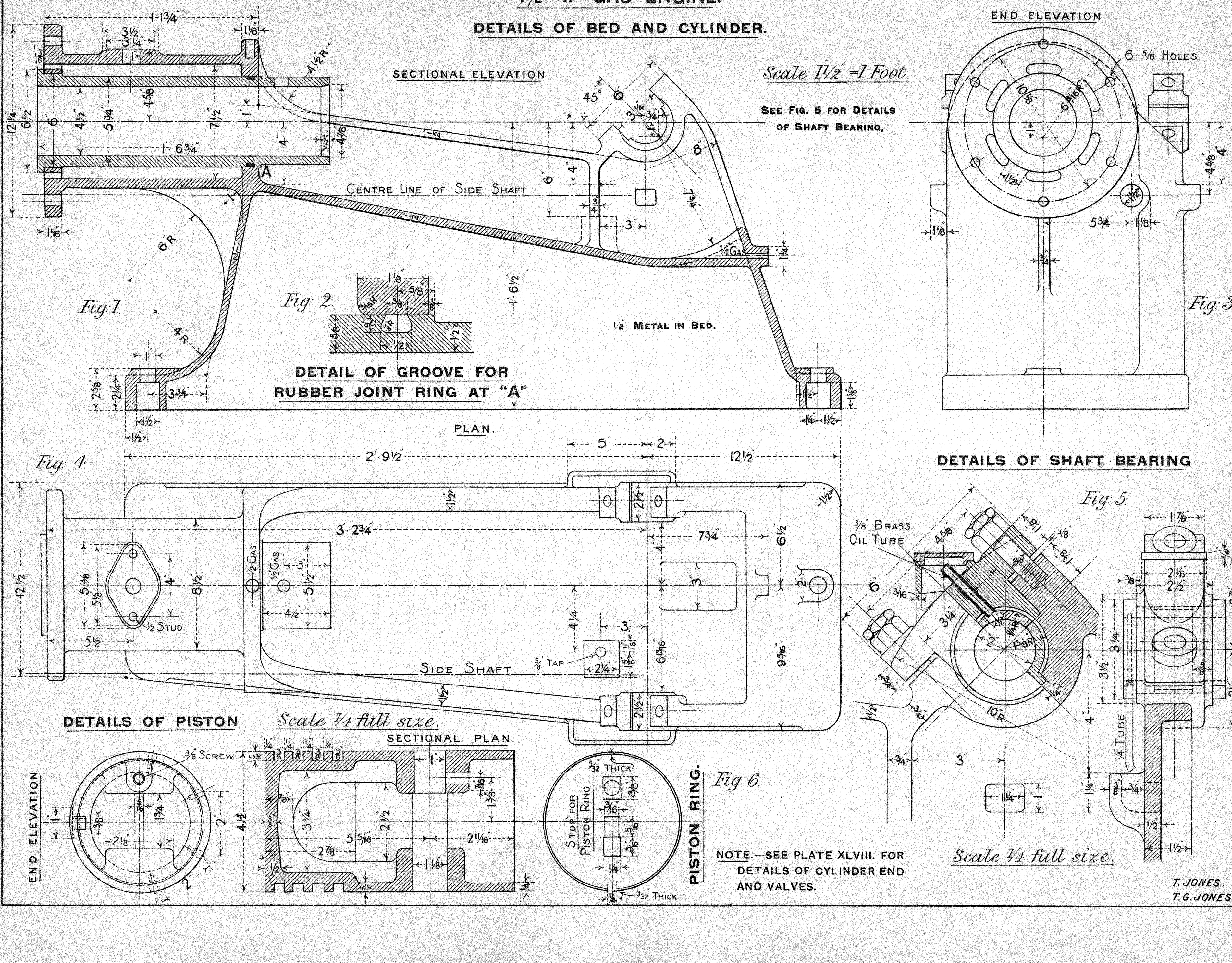
4½" × 11" GAS ENGINE.
DETAILS OF BED AND CYLINDER.
SECTIONAL ELEVATION
Scale 1½" = 1 Foot.
SEE FIG. 5 FOR DETAILS OF SHAFT BEARING,
END ELEVATION
6-⅝" HOLES
CENTRE LINE OF SIDE SHAFT
½ METAL IN BED.
¼ GAS
Fig: 1.
Fig: 2.
DETAIL OF GROOVE FOR RUBBER JOINT RING AT "A"
Fig: 3.
PLAN.
Fig: 4.
½ GAS
½ STUD
⅝ TAP
SIDE SHAFT
DETAILS OF SHAFT BEARING
Fig: 5.
⅜" BRASS OIL TUBE
¼" TUBE
Scale ¼ full size.
DETAILS OF PISTON
Scale ¼ full size.
SECTIONAL PLAN.
⅜ SCREW
END ELEVATION
5/32 THICK
STOP FOR PISTON RING
3/32 THICK
PISTON RING.
Fig. 6.
NOTE.—SEE PLATE XLVIII. FOR DETAILS OF CYLINDER END AND VALVES.
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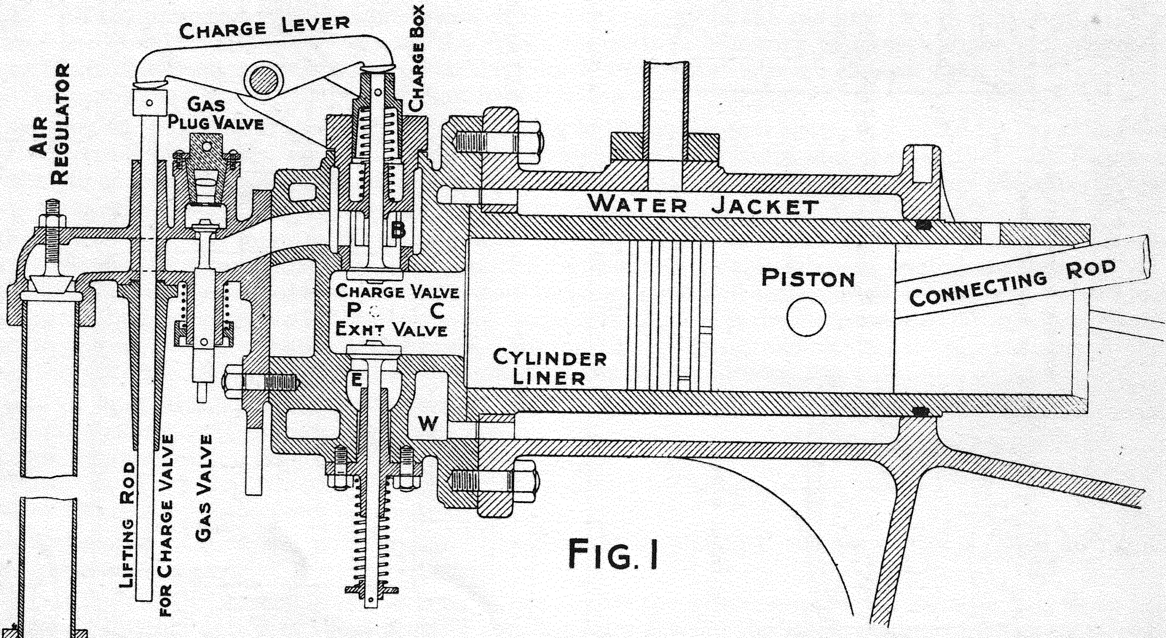
Plate XLVIII.—$4\frac{1}{2}'' \times 11''$ GAS ENGINE.

DETAILS OF CYLINDER END AND VALVES.

THE drawings give the details of the cylinder end, together with the valves for the admission and exhaust of the gases; and details of some of the other parts are shown only in the accompanying illustration, Fig. 1.

The cylinder end is secured to the back flange of the cylinder by six $\frac{5}{8}''$ studs, and is divided into four compartments, viz. :—(1) the compression space C, in communication with the cylinder; (2) the space B into which the charge box is fixed: (3) the exhaust passage E; (4) the water space W surrounding the other three compartments, and in connection with the water jacket of the cylinder.



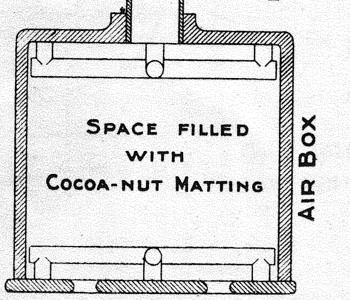


FIG. 1

Fixed to the back of the cylinder end is the gas valve box—shown only in Fig. 1—so that the passage leading from the air suction pipe and the gas valve is connected with the passage to the charge box. The pressure of gas immediately above the gas valve is regulated by the gas plug valve, the inside of which is in communication with the gas supply pipe The air required for the explosion of the gas charge enters at the bottom of the air box which contains a quantity of cocoa-nut matting, and passes the air regulator on its way to the cylinder.

The gas, charge and exhaust valves are actuated by cams (the gas valve indirectly through the governor), and hence it is necessary that the valves be held on their seatings by springs, except when the cams are acting. The governor is mounted on the lifting rod of the charge valve. The charge valve is actuated by its cam on the horizontal side shaft—referred to in the description on the previous page—through the lifting rod and the charge lever.

The side shaft runs at one-half the speed of the crank shaft, and the two valve cams are so fixed on this shaft that, during the suction stroke, the lifting rod of the charge valve and the gas valve is lifted by the charge cam—and the charge valve depressed—and a mixture of gas and air is drawn into the cylinder.

During the return compression stroke all the valves remain closed, and the explosive mixture is compressed by the piston into the compression space C, which communicates with a red-hot nickel tube by the small passage P. When the compression is complete the mixture is ignited by the heat of the tube. During the following working stroke the valves remain closed; but, on the return stroke the gases are discharged into the exhaust pipe and box by way of the passage E through the lifting of the exhaust valve by the exhaust cam.

Should the speed of the engine be too high the tripping blade of the governor misses the stem of the gas valve and so no gas is drawn into the cylinder—only air—and there is consequently no explosion and therefore no impulse during the following two revolutions.

EXERCISES.

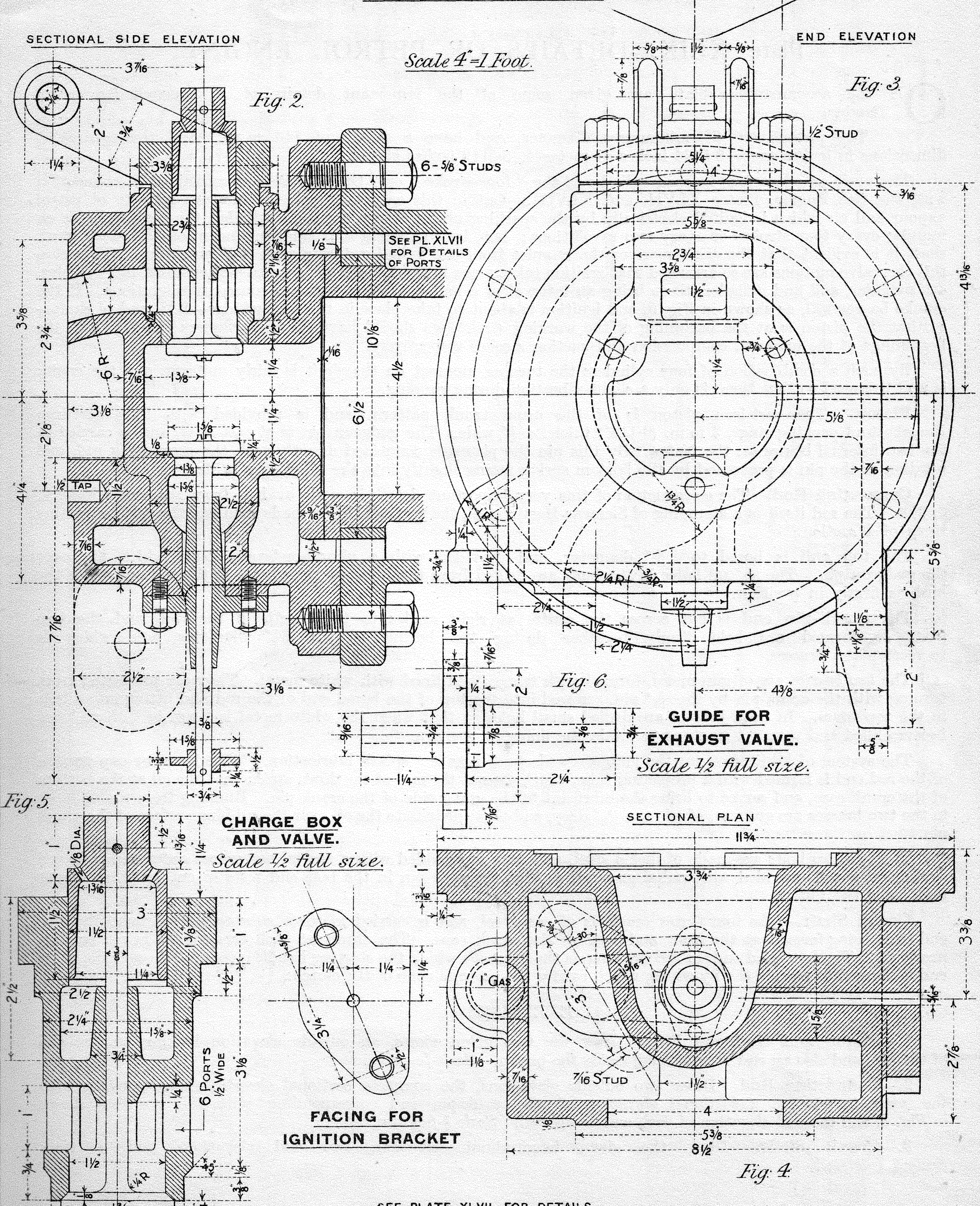
1.—**Charge Box and Valve.** Draw the sectional elevation as given in Figs. 2 and 5, an elevation as partly shown in Fig. 3, a sectional plan through the ports, and a plan. *Scale full size.*

2.—**Cylinder End.** Draw the three views as given in Figs. 2, 3 and 4, and add a plan projected from the sectional elevation. *Scale $\frac{2}{3}$ full size.*

4½" × 11" GAS ENGINE.

DETAILS OF CYLINDER END.

Scale 4" = 1 Foot.



SEE PLATE XLVII. FOR DETAILS OF ENGINE BED AND CYLINDER.

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Plate XLIX.—DETAILS OF PETROL ENGINE.

ON the accompanying plate are given some of the important details of a four-cylinder petrol engine.

The pistons are 90 millimetres diameter, and have a stroke of 140 m.m; and the equivalent dimensions in inches are :—3·5434 inches diameter × 5·5119 inches stroke.

The cycle of operations in each cylinder is the "four-stroke" or "Otto" cycle—as explained in connection with the Gas Engine, Plates XLVII. and XLVIII.,—but in this case the explosive mixture consists of petrol vapour and air which is ignited electrically by the sparking plug. Hence, for each cylinder, the shaft receives an impulse every four strokes or every two revolutions; but, by combining four single cylinders on the same crank shaft it is evident that the shaft receives the impulse from one cylinder during each half revolution. To secure this the valve motions are so arranged that ignition takes place in the following sequence—1st cylinder, 3rd cylinder, 4th cylinder, and 2nd cylinder, there being an interval of a half revolution between consecutive ignitions. If the cranks be vertical, as shown in Fig. 5, and ignition is about to take place in the 1st cylinder—L.H. end of shaft—so that the piston is at the beginning of the working stroke, we should have, in the cylinders II., III., IV., the beginnings of the exhaust, compression and suction strokes respectively.

By such a combination of four cylinders the turning moment on the shaft is fairly uniform, and the engine is well balanced so that there is only a slight vibration during running.

Piston. The cast-iron piston is of the usual trunk pattern, and is provided with four cast-iron Ramsbottom packing rings, 3 m.m. (0·118″) thick × $\frac{1}{4}$″ wide. The gudgeon pin is of cast-steel, and is carried in the two internal bosses on the piston. By this pin the piston is connected directly with the connecting rod, and rotation of the pin is prevented by the bottom spring fitting slightly into the $\frac{1}{4}$″ grooves in its ends.

Connecting Rod. The construction of this very important detail is made evident from the views Figs. 1, 2, 3, 4. The rod itself is a stamping of Siemens-Martin steel, the two ends being made solid and the part connecting them of I section.

The **top end** is bored to $1\frac{3}{16}$″ diameter, and provided with a phosphor-bronze bush which fits over the gudgeon pin. The $\frac{3}{16}$″ oil holes in the bush are drilled from the corresponding holes in the rod end when the bush is firmly in position.

The **crank-pin end** is cut across the centre at right angles to the length of the rod, and the two parts so formed are held firmly together during the boring of the $2\frac{1}{8}$″ diameter hole which is to receive the brasses.

The *two brasses* are of gun-metal, formed with flanges and lined with white metal. They are prevented from turning with the crank-pin by the $\frac{3}{16}$″ pin, screwed into the top of the bored end of the rod and fitting into a hole in the top brass. In both brasses are drilled small holes so that when the white metal is "run in" these holes become filled and thus act as stops to retain the lining in position.

The system of lubrication for the gudgeon and crank pins is "splash lubrication." The lower or cap portion of the rod end is formed with a projecting tube, which dips at the end of the down stroke into the oil at the bottom of the crank case, and serves to bring the lubricant to the under side of the crank pin. Running from the oil holes in the two brasses are grooves, $\frac{1}{8}$″ wide × $\frac{1}{32}$″ deep, and these distribute the oil over the rubbing surface of the crank pin.

The two cap bolts are made of nickel steel, and each is provided with a special nut whose outer end is turned cylindrically and provided with radial grooves, so that the split pin in the bolt end rests in one or other groove and locks the nut.

Crank Shaft.—The four-throw crank is of cast-steel, and is carried in three substantial bearings, the two end ones being formed at the front and back of the crank case. The throw of each crank is 70 m.m., or very nearly $2\frac{3}{4}$″. The left-hand end, in the drawing, is the one with which the starting handle is connected; and the other end is flanged to receive the fly-wheel, which carries the internal cone of the clutch.

EXERCISES.

1.—**Piston and Gudgeon Pin.** Draw the two given views, an outside elevation looking on the end of the pin, and also an end view looking inside the piston. *Scale full size.*

2.—**Connecting Rod.** Draw two outside elevations, the complete sectional elevation corresponding with the view in Fig. 2, a sectional elevation of the crank-pin end corresponding with the position shown in Fig. 1, and the end elevation of each end of the rod. *Scale $\frac{2}{3}$ full size.*

3.—**Crank Shaft.** Draw the given longitudinal elevation, two end elevations and a plan. *Scale $\frac{3}{8}$ full size.*

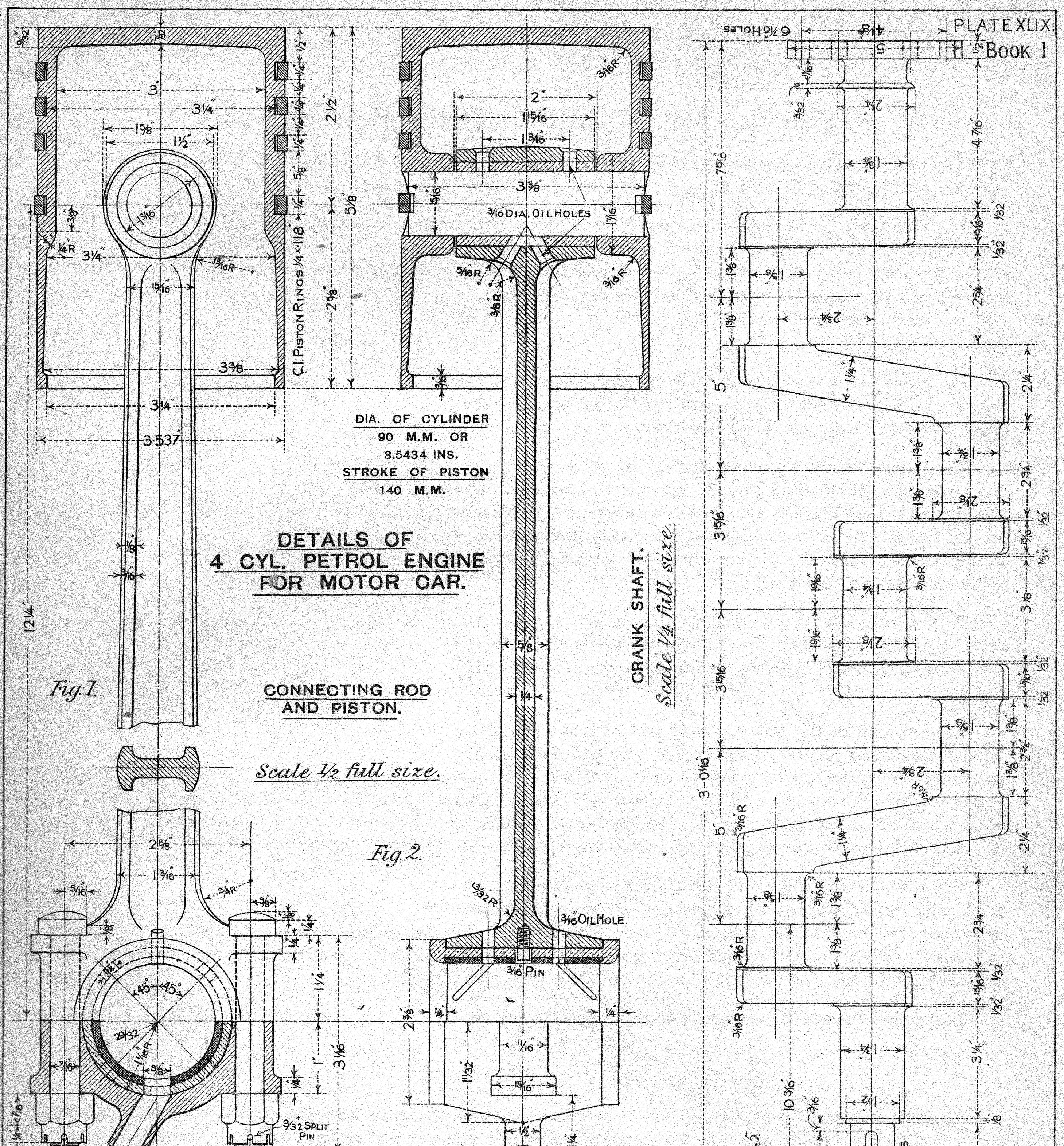
PLATE XLIX
Book I
DETAILS OF
4 CYL. PETROL ENGINE
FOR MOTOR CAR.
DIA. OF CYLINDER
90 M.M. OR
3.5434 INS.
STROKE OF PISTON
140 M.M.
CONNECTING ROD
AND PISTON.
Scale 1/2 full size.
CRANK SHAFT.
Scale 1/4 full size.
Fig. 1.
Fig. 2.
Fig. 3.
Fig. 4.
Fig. 5.
C.I. PISTON RINGS 1/4 × 1/8
3/16 DIA. OIL HOLES
3/16 OIL HOLE.
3/16 PIN
3/32 SPLIT PIN
WHITE METAL LINING 3/32 THICK
OIL GROOVES 1/8 × 1/32 DEEP
3/16 DIA.
6 7/16 HOLES
2 KEYWAYS 3/8 × 1/8
1 1/8 DIA. 20 THDS PER 1"
3/8 HOLE
1/4 TAPPED HOLE
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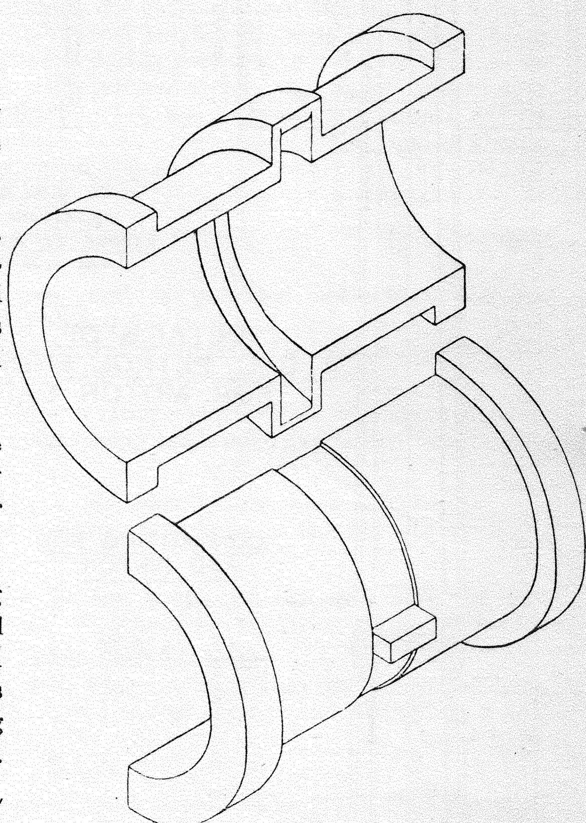
Plate L.—SELF-LUBRICATING PEDESTALS.

THE accompanying drawings represent a self lubricating pedestal, for a $3\frac{1}{2}$ inch shaft, made by Messrs. Barrett & Co., Bradford.

Self-lubricating bearings have, for many years, been universally adopted for the high-speed shaft bearings of dynamos—see the details of electrical machines in Book IV.,—but the same attention to efficient lubrication of the ordinary pedestal is not at present, general. That such a system of lubrication adds considerably to the life of a bearing and reduces the friction is beyond question; and, as shown in this example, the bearing may still be of simple design.



The exact forms of the pedestal body and brasses are—by the aid of the isometric sketches—clearly indicated, and therefore much detailed description is not necessary.

The pedestal body resembles that of an ordinary pedestal, but surrounding the bottom brass in the centre of its length is a rectangular recess R which acts as an oil reservoir. The small projection cast on the bottom brass, and fitting between snugs at the bottom of the oil reservoir, serves to prevent the rotation of the brasses with the shaft.

To accommodate the lubricating ring, which rests on the shaft, the top brass is of special design—the portion directly above the ring being of larger radius than the part on either side.

On each side of the pedestal body and cap, and projecting beyond the flanges of the brasses, is cast a trough of substantial proportions completely surrounding the shaft, so that the oil which works out from between the rubbing surfaces is collected. This oil is drawn off occasionally, and may be used again by passing it into the oil reservoir through the large hole in the top of the cap.

The lubricating ring is made of a strip of steel, $\frac{3}{8}''$ wide $\times$ $\frac{1}{16}''$ thick, with its ends formed with a hook and an eye, so that it may be sprung over the shaft and then closed, instead of having to thread it on the shaft as would be the case if the ring were solid. When the shaft rotates, the ring also rotates slowly, and since it dips into the oil in the reservoir it carries continuously to the shaft a small supply of oil.

The ratio of *length of bearing* to *diameter of shaft* is 2 to 1.

EXERCISES.

1.—**Top Brass.** Draw the outside longitudinal elevation, the cross sectional elevation through the centre of its length, the outside plan and the view looking on the inner curved surface. *Scale $\frac{3}{4}$ full size.*

2.—**Pedestal Cap.** Draw two half elevations and the corresponding half sections, the plan and the view, projected from the side elevation, looking on the inner curved surface. *Scale $\frac{3}{4}$ full size.*

3.—**Pedestal Body.** Draw the elevation and section as in Fig. 1, the plan and the half end elevation and cross section. *Scale $\frac{3}{4}$ full size.*

4.—**Complete Pedestal.** Draw Fig. 1, the plan—half complete outside view and half with cap removed—Fig. 2, and the end elevation projected from the plan. *Scale $\frac{2}{3}$ full size.*

3½″ SELF-LUBRICATING PEDESTAL.

FRONT ELEVATION — SECTIONAL ELEVATION.

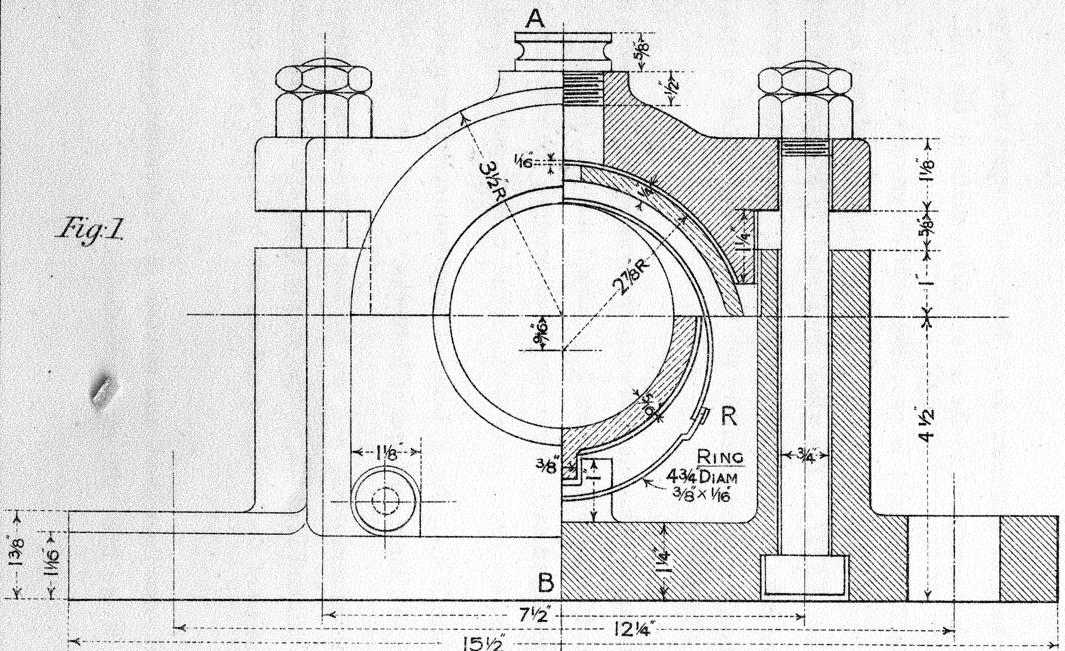


Fig. 1.

SECTIONAL ELEVATION AT A. B.

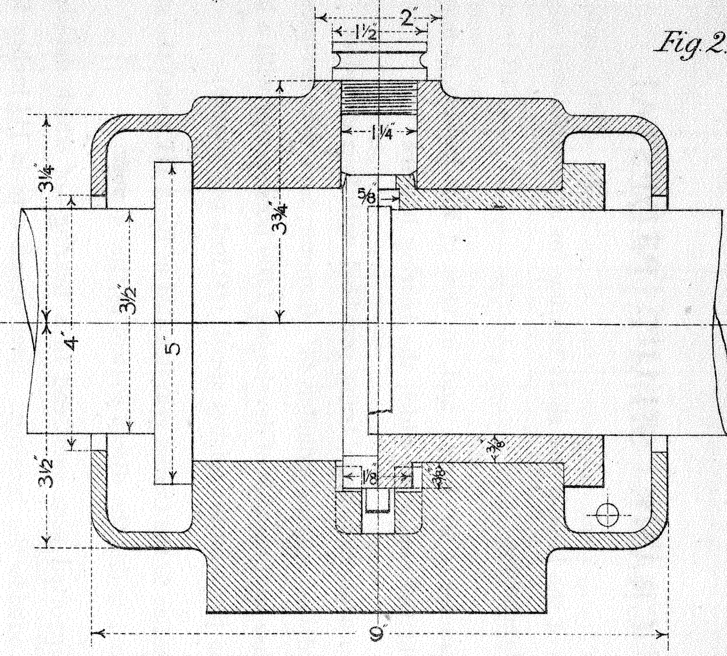


Fig. 2.

PLAN WITH CAP REMOVED. — PLAN OF PEDESTAL BODY.

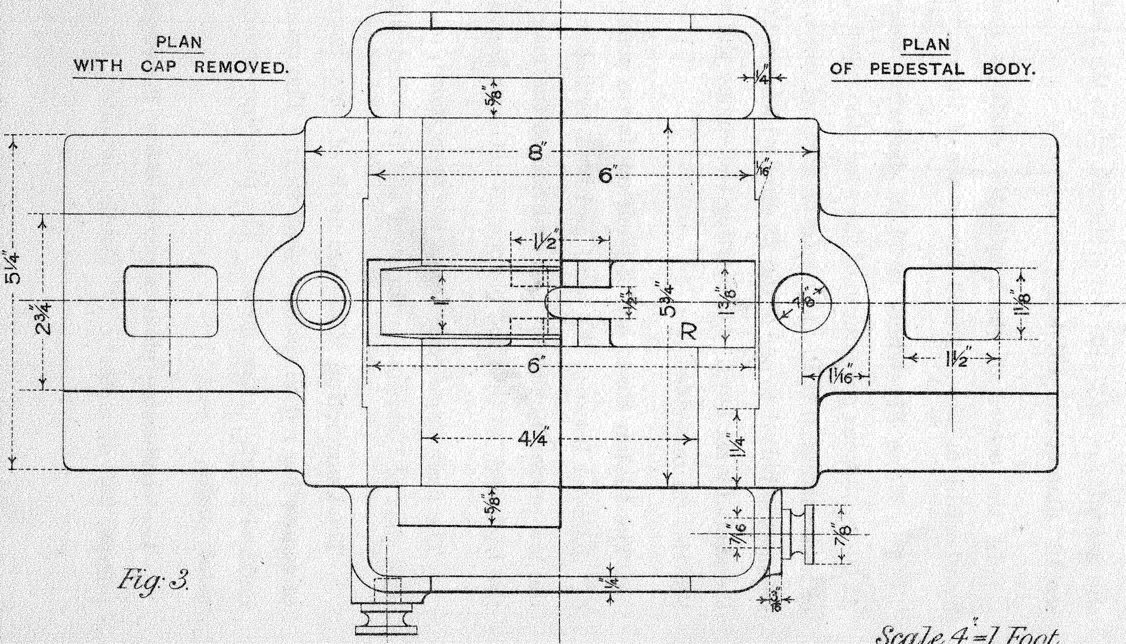
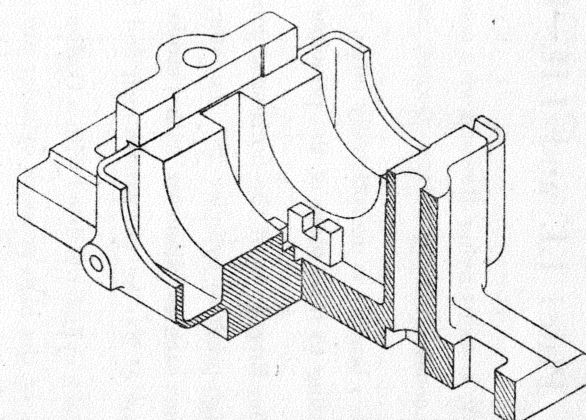


Fig. 3.

ISOMETRIC VIEW OF PEDESTAL BODY.



THIS VIEW MUST NOT BE DRAWN.

Scale 4″ = 1 Foot

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Plates LI. and LII.—LARGE CRANK SHAFT PEDESTAL.

THE accompanying drawings give complete details of the large crank shaft bearing for a powerful mill engine, constructed by Messrs. George Saxon Ltd., Manchester.

This bearing is exceptionally large, being $18\frac{1}{2}''$ diameter $\times$ $40''$ long; and, in view of this feature, it may be interesting to note some of the principal dimensions of the engine to which the bearing belongs.

The **Engine** is of the horizontal condensing triple expansion type, being provided with four cylinders, viz., one high-pressure cylinder 28″ diameter, one intermediate-pressure cylinder 44″ diameter, and two low-pressure cylinders each 48″ diameter, with a stroke of 5 ft. 6 inches, and capable of developing 2,500 horse-power with a boiler pressure of 200 lbs. per square inch and 60 revolutions per minute; (piston speed, 660 feet per minute).

Each cylinder is fitted with a Corliss valve gear (similar to the Corliss gear detailed in Book III.), and the steam valves of the high-pressure cylinder are coupled to, and under the direct control of, the governor.

The **Crank-shaft** is of Siemens-Martin steel, supported near each end in a bearing—of which details are given—and fitted with two overhanging cranks just outside the bearings. Each crank arm is $11\frac{1}{2}$ inches thick, and has a throw of 2′—9″: it is shrunk on to the shaft, and further secured by pins and steel keys. The crank pins are 12″ diameter $\times$ 13″ long.

In the centre of its length the shaft is increased in diameter to 2 feet, and carries the large rope fly-wheel by means of which the power of the engine is transmitted to the several main shafts in the mill.

The **Rope Fly-wheel** is 28 ft. diameter, weighs 73 tons, and is provided with 50 grooves for $1\frac{5}{8}''$ ropes. At 60 revolutions per minute of the wheel the ropes have a velocity of exactly 1 mile per minute, and each is capable of transmitting 50 H.P. (*See notes on power of ropes, plate XXIII.*) The wheel rim is made in 28 segments, and each segment is bolted to one of the 28 arms which are fitted into the two bosses of the wheel. Each boss is secured to the shaft by six keys. On starting the engine the wheel is barred round by the barring rack, which is cast inside the rim on one side.

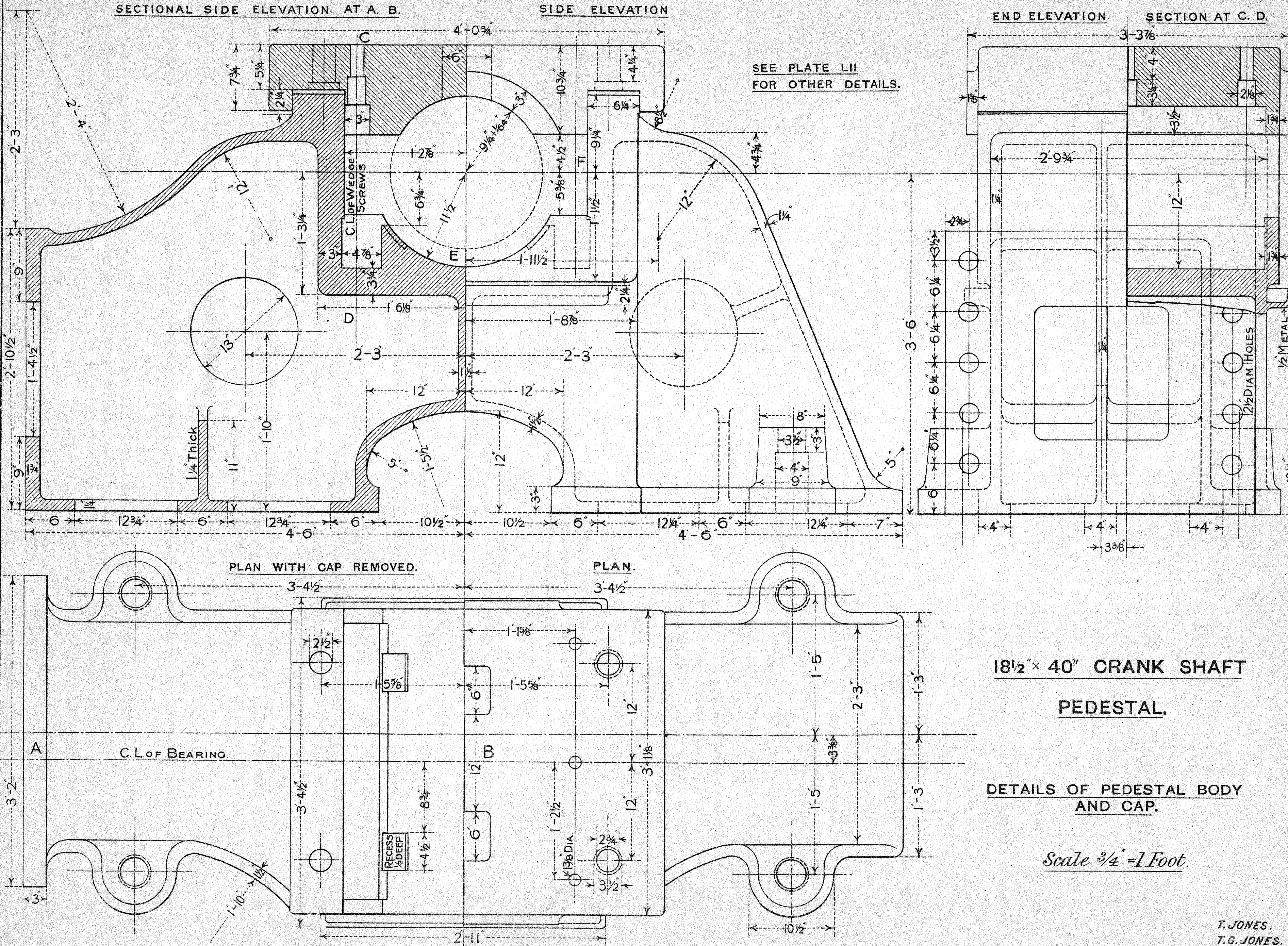
The **Piston Rods** are of Siemens-Martin steel, being $7\frac{1}{2}''$ diameter between the crosshead of the first piston, $6\frac{1}{2}''$ diameter between the front and back cylinders, and 5″ diameter where it is carried through the back cover of the low-pressure cylinder. In addition to the main crosshead each rod is supported between the cylinders and at the back end, so that the pistons do not bear heavily on the lower parts of the cylinders and cause unequal wear.

The main crossheads are of the type shown on Plate XXX., Fig. 2, being of forge hammered scrap iron, fitted with pins of Siemens-Martin steel and provided with two large cast-iron slide blocks.

The **Connecting Rods** are made of forge hammered scrap iron, and are 16′ long between the centres (ratio of connecting rod to crank = 5·82 to 1). The crank pin end is of the marine engine type, and the brass steps are lined with anti-friction metal. The phosphor-bronze steps of the crosshead end are held by a strap with cotter and safety bolts.

The condensing plant is placed below the engine room floor level, and consists of jet condensers and two air pumps, 38″ diameter $\times$ 19″ stroke, actuated by L levers from the main crossheads.

Continued.



18½" × 40" CRANK SHAFT

PEDESTAL.

DETAILS OF PEDESTAL BODY AND CAP.

Scale ¾" = 1 Foot.

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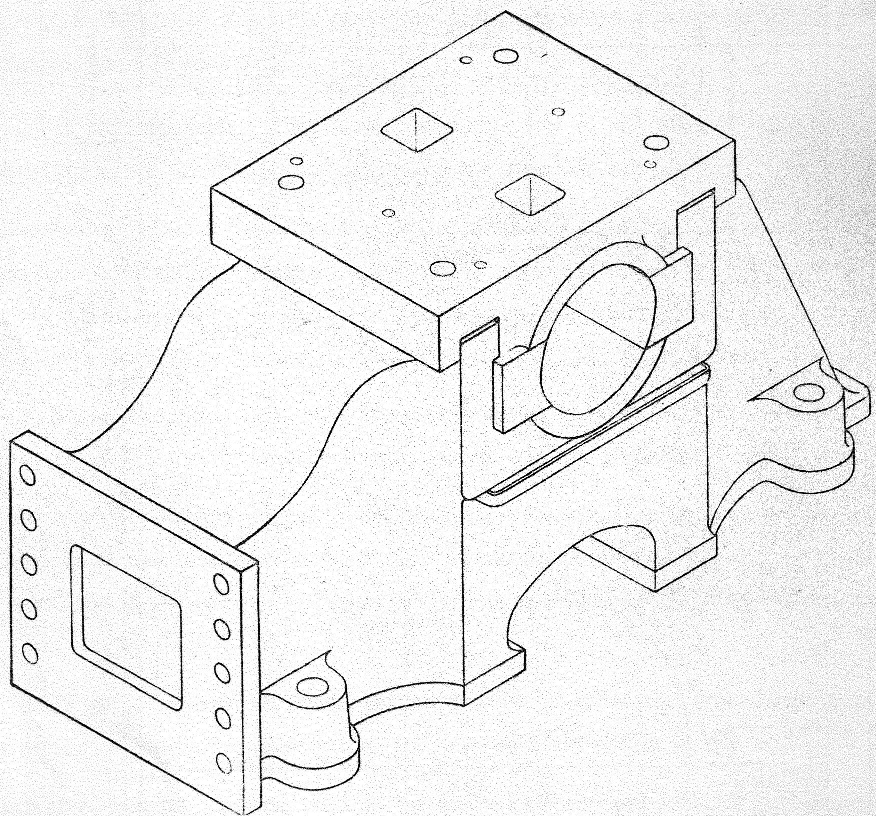
ENGINE BEARING.

The general appearance of the bearing is indicated by the accompanying isometric sketch.



The pedestal body is of box section, of $1\frac{1}{4}''$ metal, stiffened inside by substantial webs. It is 9 ft. long, and bolted to the foundation by four $3\frac{1}{2}$ inch bolts. At one end is a vertical flange, $2'$—$10\frac{1}{2}''$ high $\times$ $3'$—$2''$ wide, by which it is bolted to the end of the engine frame. The portion which receives the steps is of rectangular box form, and contains the three parts which support the shaft.

The bottom and two side steps are made of cast-iron, and lined completely with "Magnolia" metal to a thickness of $\frac{1}{2}$ inch. To retain the lining in position the curved surfaces of the steps have dovetailed grooves cut in them.

The **bottom step**—see Fig. 1, Plate LII.—is carried in the curved face E of the pedestal body, and its bearing surface subtends an angle of 110° at the centre of the shaft. On the under side of this step are cast four rectangular snugs which fit into rectangular recesses in the face E.

Each **side step**—Fig. 2, Plate LII.—subtends an angle of 65° at the centre of the shaft, and is formed with flanges at each side. The step is held against the shaft by a cast-iron wedge Fig. 3, which is supported from the cap by three screws—Fig. 4—and bears against the vertical face F of the pedestal body. On the wedge there is a slope of six vertical to one horizontal, and therefore for one turn of each of the three adjusting screws the side step would be fed inwards a distance equal to $\frac{1}{6}$ of the pitch of the screws, *i.e.*, $\frac{1}{6}$ of $\frac{1}{6}''$ or $\frac{1}{36}$ inch.

The bearing is so designed that, by unscrewing the adjusting screws from the wedge, and allowing the latter to fall into its lowest possible position, the side step can be moved round on the shaft and removed—after the cap has been raised—without disturbing the shaft.

The substantial cap is fixed to the body by four $2\frac{1}{2}''$ studs, and the part against the shaft is bored to a diameter $\frac{1}{32}''$ larger than the shaft It is provided with two large holes to give easy access to the shaft and to form oil reservoirs.

An oil trough is cast on each side of the pedestal body immediately below the steps.

EXERCISES.

1.—**Pedestal Cap.** Draw the complete side and end elevations; the plan; and, projected from the plan, the half sectional elevation through the centre and the half sectional elevation through the holes of the adjusting screws. Add, also, a view, projected from the side elevation, looking on the under side of the cap. *Scale* $2'' = 1$ *foot.*

2.—**Bottom Step.** Draw the two given views; and add the side elevation projected from the plan; the sectional side elevation through the centre, projected from the end elevation; and the view of the under side of the step, projected from the side elevation. *Scale* $2'' = 1$ *foot.*

3.—**Side Step.** Draw the end elevation, the given views and the back elevation projected from the plan. *Scale* $2'' = 1$ *foot.*

4.—**Wedge.** Draw the given views, and add a cross section through the axis of a screw, and the plan. *Scale* $3'' = 1$ *foot.*

5.—**Complete Pedestal.** Draw the side and sectional side elevations; the complete end elevation to the right of the side elevation; the plan, one half showing the outside view and the other half a section through the centre of the shaft; and the cross sectional elevation through the axis of the shaft, projected to the right from the plan. Show the steps, studs, screws, etc. *Scale* $1'' = 1$ *foot.*

If drawn on paper of Imperial size use the Scale $1\frac{1}{2}'' = 1$ *foot.*

DETAILS OF 18½"× 40" CRANK SHAFT PEDESTAL.

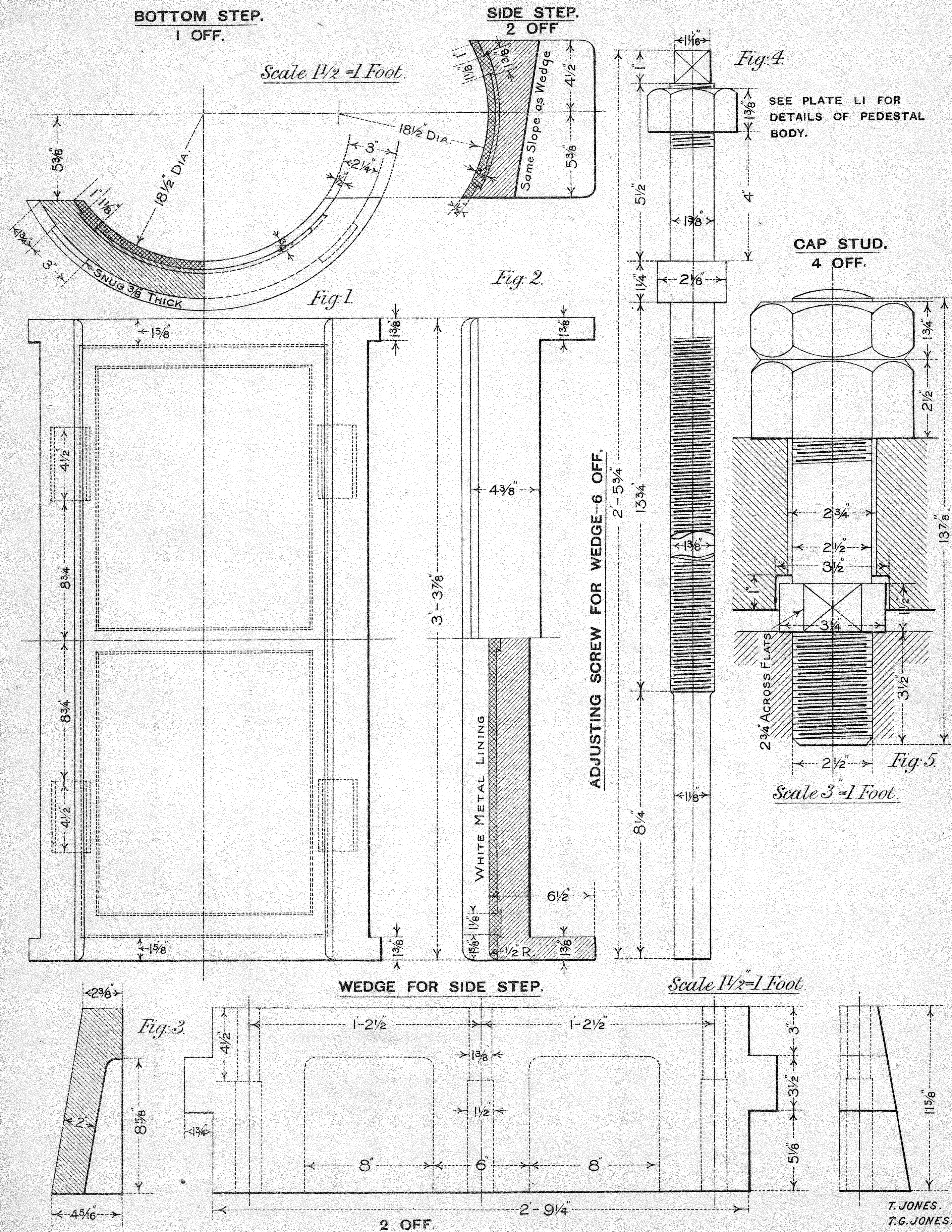


Plate LIII.—16″ JAW CHUCK FOR LATHE.

A JAW chuck is used for the purpose of holding or *chucking* work in a lathe, either for turning or boring. It has the advantage of enabling the workman to fix the work easily and quickly, and admits of adjustment when the part operated upon is not concentric with the outer portion by which it is held. The chuck is made in the form of a face plate with sliding jaws, each capable of radial motion. The jaws are stepped, so that work of various diameters can be held by moving them a short distance.

The drawing on the opposite page gives a portion of each of four views of a jaw chuck 16″ diameter. The body A is made of cast-iron, and the jaws B, the screw C and the nuts D, of steel.

The screws which are turned by the box key, have left-hand threads cut upon them. The force due to the holding of the work in the jaws is taken up by the steel collars E, which are held in position by round steel pins.

The shanks of the jaws B slide in radial grooves, and are tapped for the screws to work through, and the jaws are locked in any desired position by the steel nuts and washers.

EXERCISES.

1.—Make complete working drawings, *full size* of the following **details of the jaw chuck.** (1) A cast steel jaw, with locking nut at the back. (2) One of the screws. (3) The box key.

2.—**Jaw Chuck.** Draw and complete the various views given, and add a plan. *Scale* 6″ = 1 *foot.*

16" JAW CHUCK.

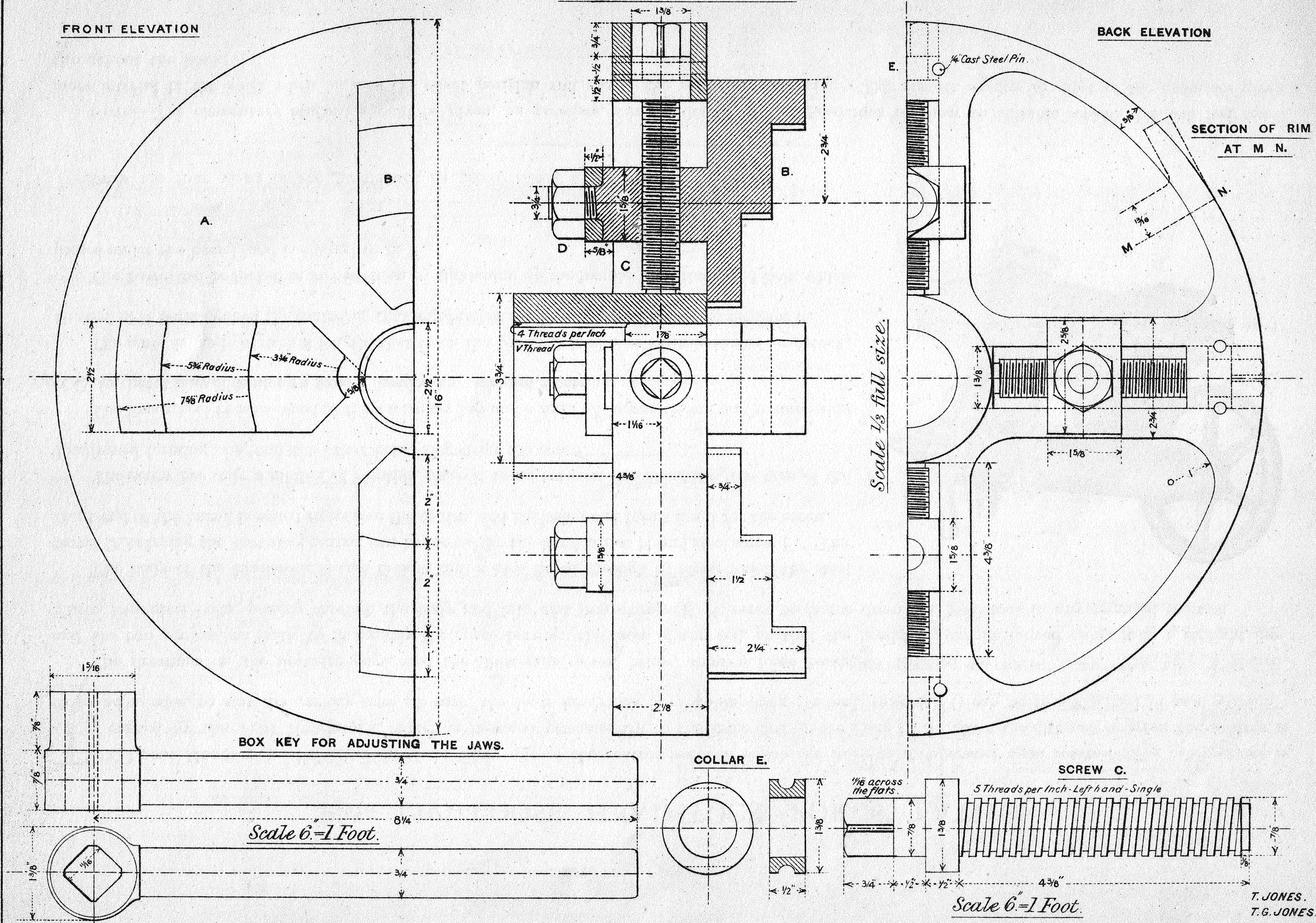


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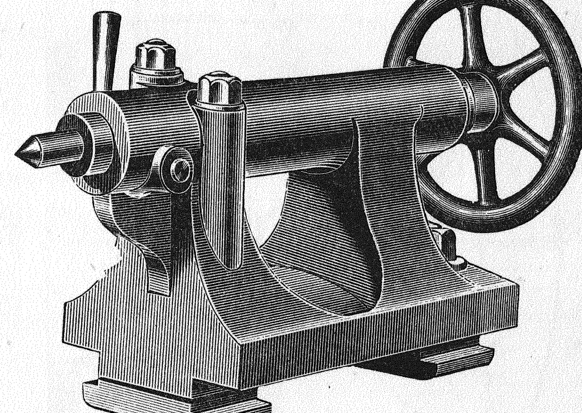
Plate LIV.—LOOSE HEADSTOCK FOR 8″ LATHE.

THE **Loose Headstock** of a lathe serves to carry one of the centres between which the work to be operated upon rotates. The other centre is carried by the **Fast Headstock,** which is fastened permanently to the lathe bed. (See Plate LV.) Since the distance between the centres is to be adjustable, to suit the various sizes of work, the loose headstock is movable along the bed, to which it can be firmly bolted in any position.

The drawings on the opposite page, and the illustration given below, show a loose headstock as used for heavy work. The base is planed, and the two projections made to fit exactly the space between the faces of the bed, so that the headstock can be moved along it in a straight line. Three long steel bolts, passing through the body and into cast iron clamps K K, serve to fasten down the headstock in any required position.

The body of the headstock is cast hollow, and a hole bored through it, along which the steel barrel D, carrying the cast steel centre, can be moved by the hand wheel H and steel screw F. The front end of the barrel is coned to receive the centre, and the other end forms a nut for the screw.



The screw has only a motion of rotation, since it turns between two shoulders, the boss of the handwheel forming one, and the other being forged on the screw.

The handwheel H is fastened to F by a square key and a nut. A keyway is cut on the underside of D, so that it may slide along a fixed T headed key, and not rotate.

The screw is made of such a length, that when the barrel D is almost entirely in the headstock, the end of it butts against the centre, so that any further rotation of the screw would displace it.

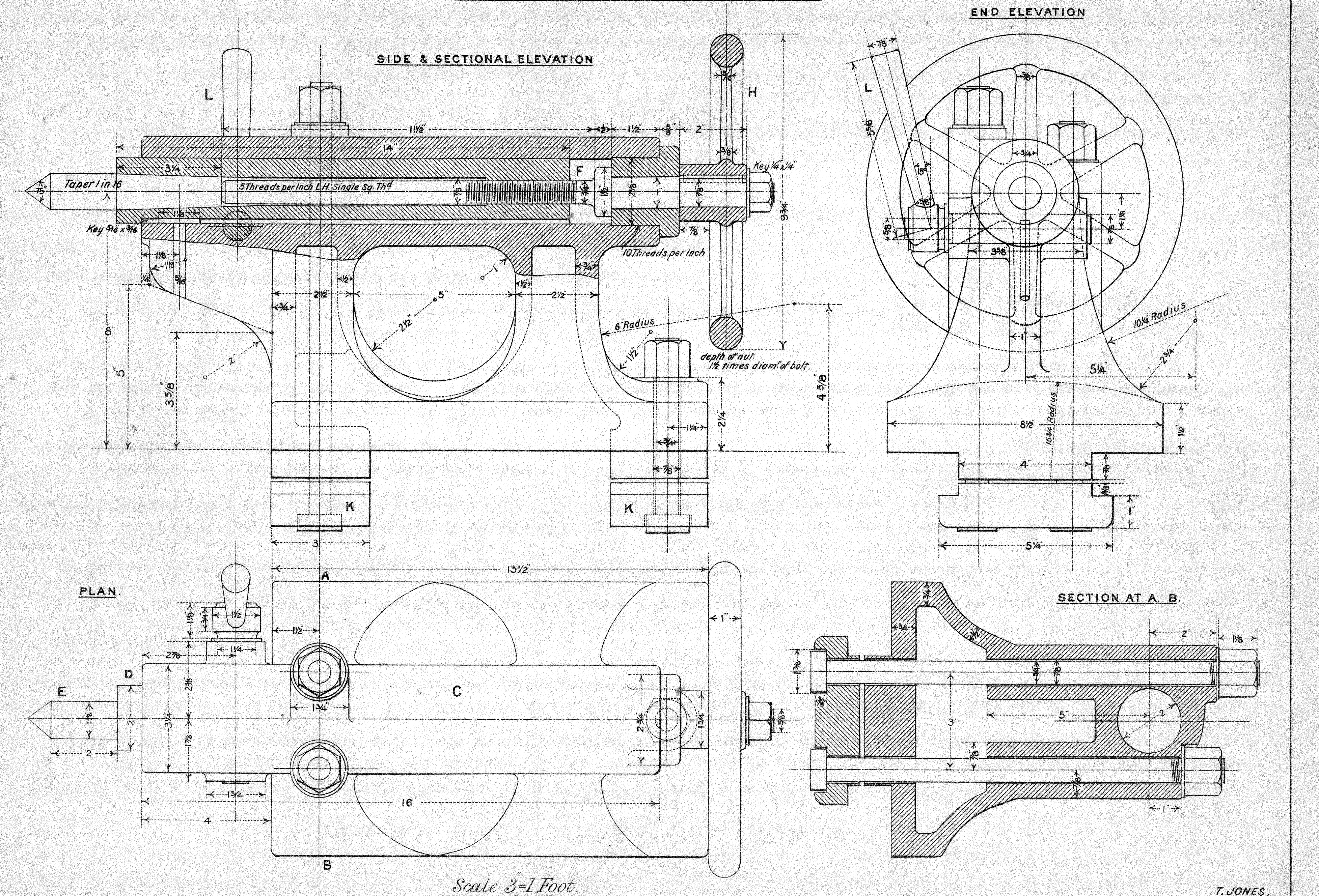
The barrel can be locked in any position by tightening up the handle L on the round bolt, which passes under the barrel, and is shaped to fit it.

EXERCISE.

Draw the four views of the **headstock** as given. *Scale ½ full size.*

NOTE.—The elementary student should be given, as exercises, various details of this headstock to draw to suitable scales. He will feel much more interest in the work when he sees the exact position and use of the piece he is drawing. This remark applies to most of the examples given throughout the book.

LOOSE HEADSTOCK FOR 8" LATHE.



Scale 3"=1 Foot.

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Plate LV.—FAST HEADSTOCK FOR 8″ LATHE.

FIGS. 1, 2, 3 show a **back geared fast headstock** for an 8″ lathe, and **Figs. 4, 5, 6** give enlarged details of some of its parts.

The base of the body E is planed and provided with two projections, which fit exactly the groove in the bed, and thus keep the spindle H parallel with the top and sides of it. It is secured by four screws which pass into clamping pieces on the underside of the bed.

The steel spindle H is supported in the headstock by two conical brass bushes. The front bush is fitted tightly into the hole, while the other one is held lengthways by two thin circular nuts M M. An adjustment for the wear of the spindle and the bushes can be made by the nuts M M, and lock nuts N N. Conical bearings have an advantage over cylindrical ones, since with the former the centre of the spindle always remains at the same height above the lathe bed.

The end thrust on the spindle is transmitted through the washers P to the cross bar K, which is fixed on the ends of two pillars by nuts.

The cone pulley F, to which the pinion C is fastened, runs loose on the spindle, but when the wheels on the back shaft are out of gear with the wheels C and A, it is secured to the wheel A by means of a bolt whose head fits between snugs on the pulley plate—See Figs. 1 and 5. The cone plate is secured to the pulley by six $\frac{3}{8}$″ screws. The front end of the spindle has a conical hole bored in it to receive the cast steel centre, which is carefully fitted to the hole, and the end afterwards turned up in its place when the lathe is complete.

In plain bearings, in the sides of the headstock, a shaft L is placed parallel to H, upon which revolves a hollow cast-iron shaft, having keyed to its ends the spur wheel B and the pinion D.

B and D can be put in or out of gear with C and A respectively, by turning the shaft L through half a revolution, since its ends are eccentric with the portion upon which B and D revolve. A collar is pinned on the right hand end of L, and is fitted with two small handles, as shown in Fig. 6, by means of which L is rotated. A segment, cast on the arm of the headstock, prevents the handles being moved through more than 180°.

By using the back gearing—F and A being disconnected—the speed of the spindle is reduced in the ratio $\left[\frac{C \times D}{A \times B} = \frac{18 \times 18}{54 \times 54} = \right] \frac{1}{9}$, assuming that the driving belt is not moved from one pulley to another.

EXERCISE.

Draw the three given views of the **fast headstock** and add another end elevation. *Scale* 3″ = 1 *foot.*

QUESTIONS.

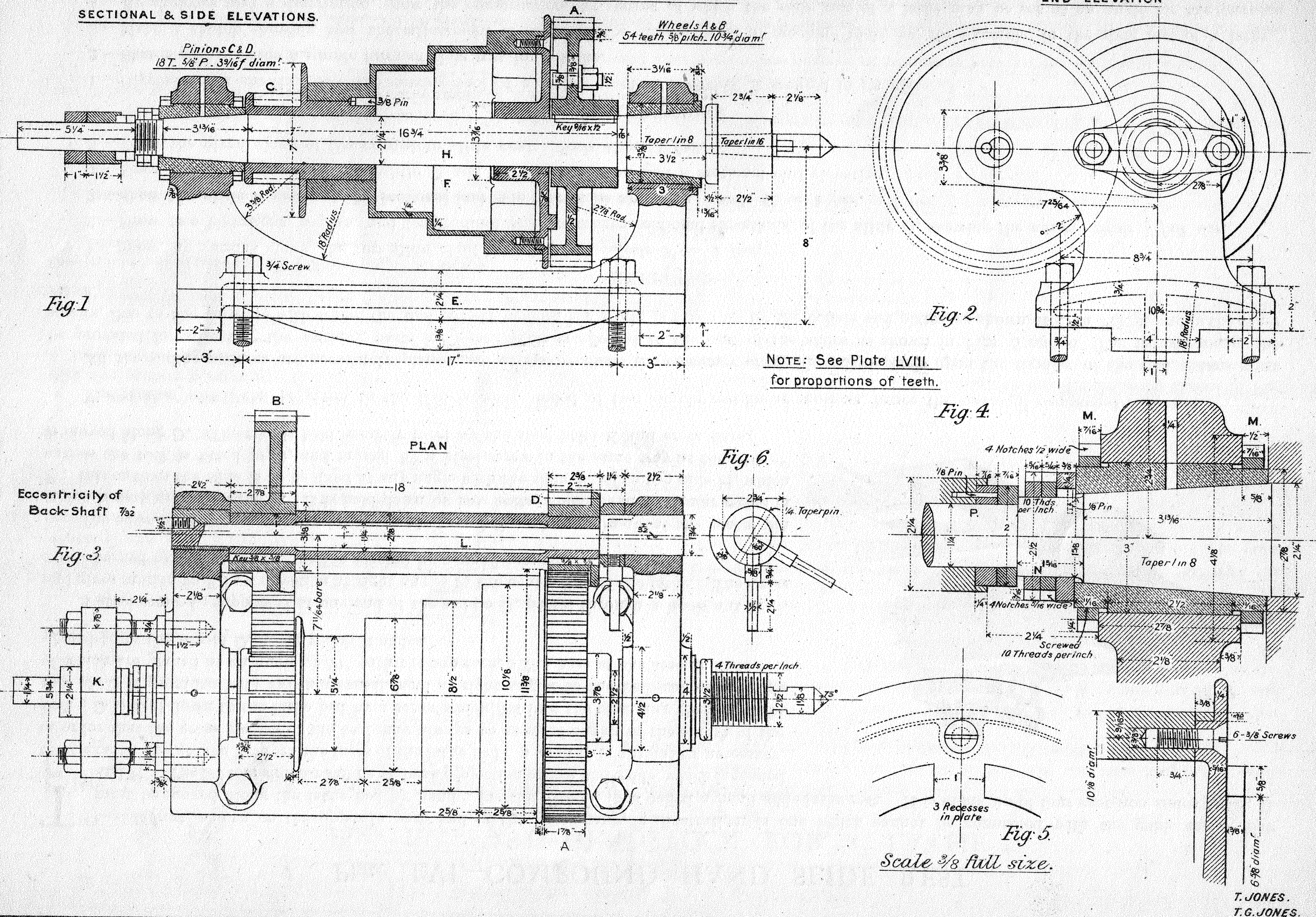
1.—Suppose the cone pulley on the headstock to be driven from one of the same size on a counter-shaft making 150 revolutions per minute, determine the various speeds of the spindle which can be obtained with and without back gearing.

2.—Give sketches, showing how you would grip and drive a round iron bar for the purpose of turning it between the centres of a lathe

NOTE.—the elementary student should be given, as exercises, various details of this headstock to draw to suitable scales. He will feel much more interest in the work when he sees the exact position and use of the piece he is drawing. This remark applies to most of the examples given throughout the book.

FAST HEAD STOCK FOR 8" LATHE.

Scale 1/6 full size.



NOTE: See Plate LVIII. for proportions of teeth.

Scale 3/8 full size.

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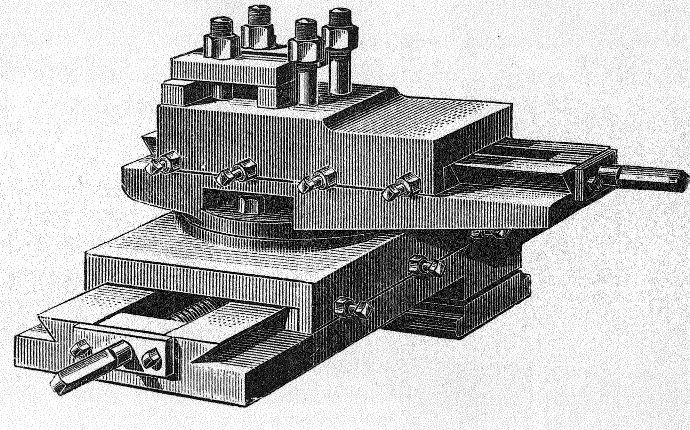
Plate LVI.—COMPOUND HAND SLIDE REST.

THE slide rest, shown on the opposite page and in the accompanying illustration, is one which cannot be connected with the guide screw, but must be moved along the lathe bed by hand: for this reason, it is called a hand slide lathe rest. It is made up of four cast-iron slides D, F, G and H. The bottom slide, or saddle D, has a projection on the underside, which is planed to fit exactly the groove between the faces of the lathe bed: a good fit is absolutely necessary, in order that the cross motion of the tool may always be at right angles to the length of the bed. D is held down to the lathe bed by a screw which fits into the clamping piece E. The top of D is provided with two horizontal faces at right angles to the bed, the outer edges of which are planed at an angle of 50° with the top face. The base of the second slide F is planed to fit the top of D, along which it slides.



A steel screw L, supported at one end of the saddle D, passes through a brass nut in F, and gives motion to F in a direction at right angles to the centre line of the lathe. The brass nut is turned to fit a hole in F, and is secured by a small screw.

The slide G rests on a circular facing on the top of F, and can turn about the centre pin which is cast with the slide F. It is held down by two bolts, which fit in a circular recess in F; this enables the slide G to be fixed at any angle with the slide F. The top slide H, which carries the tool, is fitted to G, and moved by a steel screw in the same way as the slide F is moved along D. The tool is held securely to H by the steel bolts K and cross bars.

The motion which can be given to the tool is compounded of two simple rectilinear motions, hence the name of compound slide rest.

All the sliding surfaces are accurately planed and scraped. Since the efficiency of a slide rest depends upon the rigidity of the tool, means must be provided for adjusting the working parts for wear. This is effected in the case of the slides as shown in Figs 5 and 6. The thrusts along the screws, due to the pressure with which the tool is held against the work, is taken up by the collars and plates, as shown at C in Fig 4, and in the other views.

EXERCISES.

1.—Show, by various views, the **top slide** H of the slide rest. *Scale* 9″ = 1 *foot*.

2.—Draw two elevations, a plan, and longitudinal and transverse sectional elevations, of the **slide** G, showing the screw. *Scale* $\frac{1}{2}$ *full size*.

3.—Show the **slide** F and nut, by sectional and side elevations and plan. *Scale* 6″ = 1 *foot*.

4.—Draw the given views of the **saddle** D and the screw L, and add a sectional end elevation. *Scale* $\frac{1}{2}$ *full size*.

5.—Draw the given views of the **complete slide rest**. *Scale* 4″ = 1 *foot*.

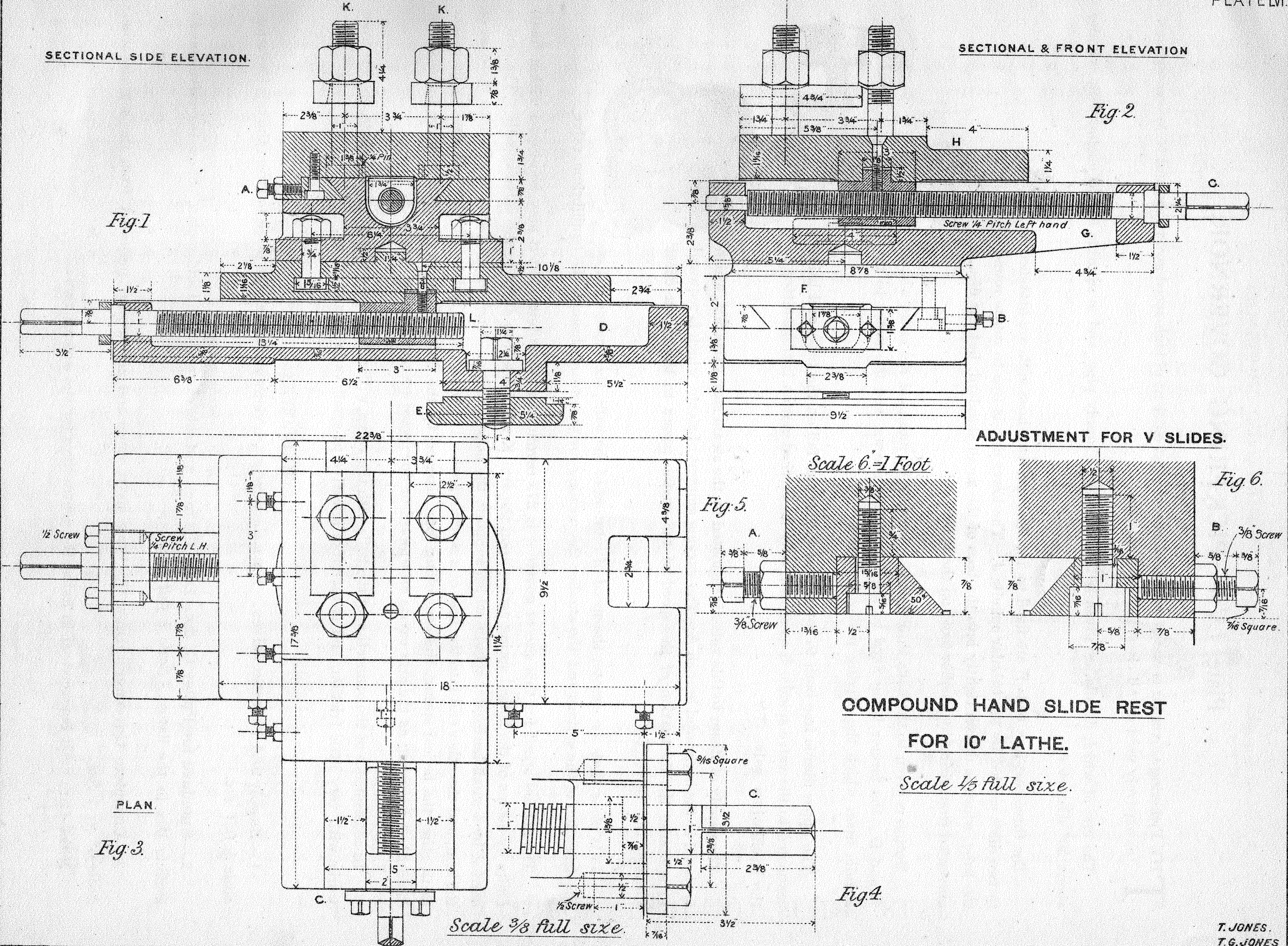
QUESTIONS.

1.—Give sketches showing how the cutting tool of a lathe, or other machine, is secured in its place.

2.—Sketch and describe a simple form of slide rest for a lathe.

3.—Make a sketch showing how the adjustment is made in the sliding parts of machine tools, as, for example, in the slide rest of a lathe.

4.—By sketches and a description, show the construction, by means of which the slide rest of a lathe may be set at an angle for the purpose of turning a taper rod. Explain why the cutting edge of the tool should be placed at the exact height of the axis of rotation, in order to form a portion of a true cone.



COMPOUND HAND SLIDE REST

FOR 10" LATHE.

Scale 1/5 full size.

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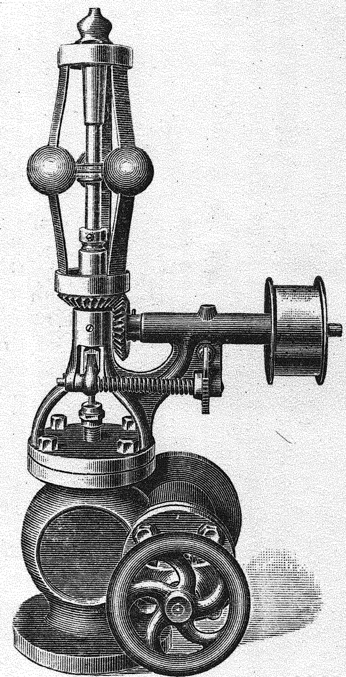
Plate LVII.—PICKERING GOVERNOR.

THE Pickering Governor is one of the simplest and most efficient of high-speed governors, and is, in consequence, extensively used on all classes of land engines.

The one of which details are given is for a 6″ steam supply pipe, and is suitable for an engine with cylinder about 30″ diameter.



The three cast-iron balls F are fixed to flat steel springs whose ends are attached to two discs carried by the tubular spindle J. The lower disc is free to rotate round the tube, which is driven tightly into the boss of the small standard, but the disc is prevented from rising by a collar fixed on the tube.

Motion is given to the balls through the lower disc by means of the mitre wheels E and D; the former being fixed to the disc by set screws, and the latter to the spindle upon which the driving pulley is keyed.

The upper disc is free both to rotate and to rise and fall on the tube, and being connected to the valve spindle S as shown, only its vertical motion is transmitted to the equilibrium valve TV.

The supply of steam to this valve is regulated by the steam stop valve SV, which is placed in the horizontal branch A. The varying amount of steam which is required to minimise the fluctuation of the speed of the engine passes to the steam chest through the valve TV.

It is clear that the greater the speed of rotation of the balls, the greater will be the deflection of the springs and the smaller the opening for the steam through the equilibrium valve.

The normal speed for this size of governor is 275 revolutions per minute, but it may be altered within small limits by means of the helical spring acting upon the arm K which fits into a slot in the valve spindle. The arm is carried by a small horizontal spindle which supports the helical spring. One end of the spring fits into the boss of the arm, and the other end into the boss of a worm wheel. The worm wheel is driven by a worm, and by its action in twisting the helical spring, the controlling force exerted by the flat springs on the balls may be increased or diminished according as the arm exerts an upward or downward pressure on the spindle.

In consequence of the connection of the balls with the valve, without the intervention of any joints, the governor works with very little friction, and is therefore sensitive to small variations of speed.

EXERCISES.

1.—**Supply and Throttle Valves.** Draw the three given views and add a plan projected from the side elevation. *Scale* $\frac{1}{3}$ *full size.*

2.—**Standard for Pulley C, &c.** Draw the front and side elevations and add a plan. Show the valve spindle stuffing box in the flange. *Scale* $\frac{1}{2}$ *full size.*

3.—**Complete Governor.** Draw the three given views and add a plan projected from the side elevation. *Scale* 3″ = 1 *foot.*

N.B.—Care must be exercised in fixing the centre lines for the several views.

PICKERING GOVERNOR FOR 6" STEAM PIPE.

Scale 1½" = 1 Foot

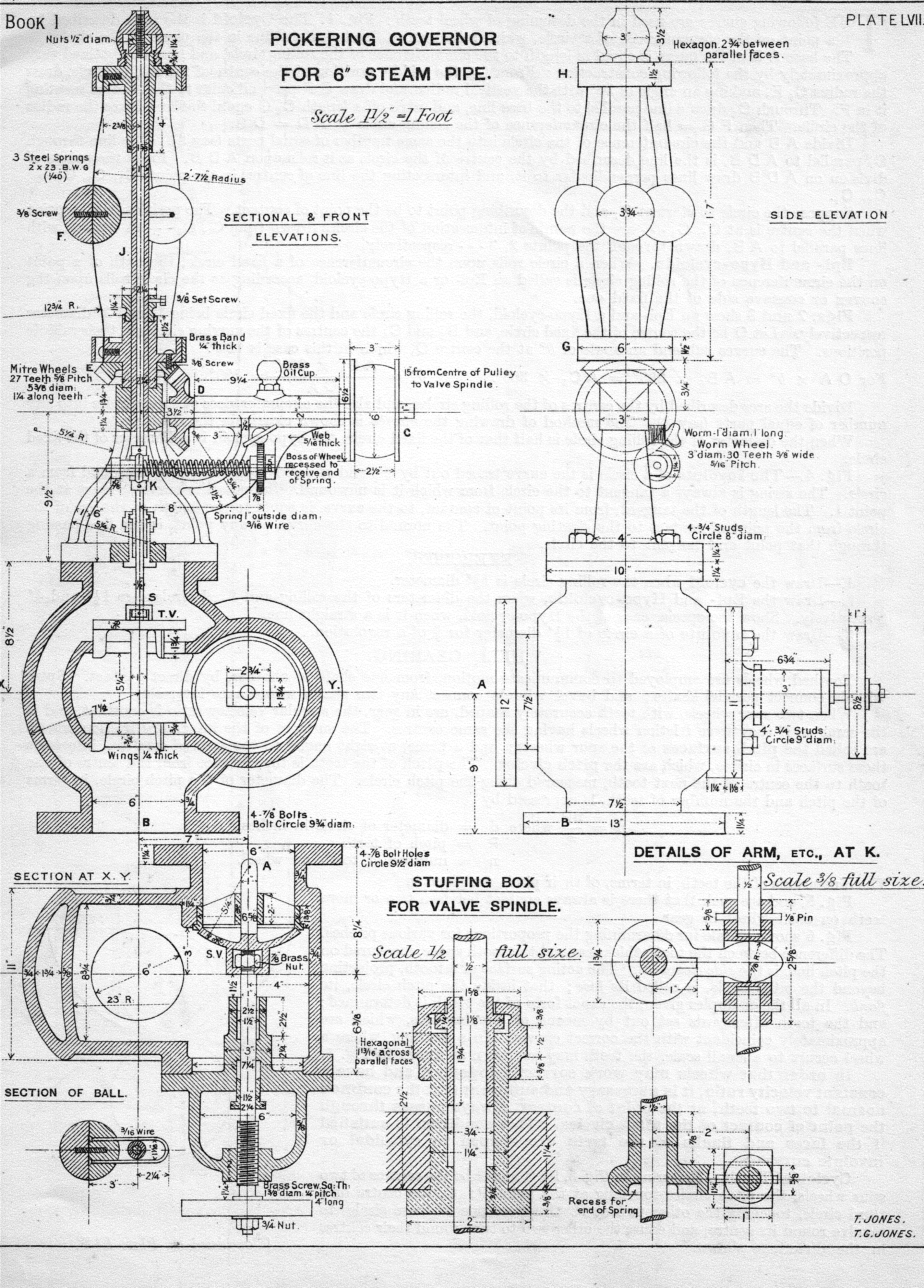


Plate LVIII.—CYCLOIDAL AND INVOLUTE CURVES.

THE following curves are used in the designing of wheel teeth: **Fig. 1. The Cycloid** is the path described by a point on the circumference of a circle, which rolls upon a fixed straight line in the plane of the circle.

The base A B of the curve is equal in length to the circumference of the rolling circle, and may be obtained very approximately by the following construction: From C_5, the middle position of the centre of the rolling circle, draw the radius C_5 E, making an angle of 30° with the vertical radius, and draw E F perpendicular to that radius, meeting it in F. Through C_5 draw a line parallel to the base line, and mark off a length C_5 G equal to three times the radius of the circle. Then F G = half the circumference of the rolling circle = A D = D B.

Divide A B and the circumference of the circle into the same number of equal parts (say 8). The line through C, parallel to A D B, is the line described by the centre of the circle as it rolls upon A D B. From the points of division on A D B draw lines perpendicular to it, and intersecting the line of centres in the points C_1, C_2, C_3 - - - C_8, C_1.

Suppose the circle to start at C_1 and the describing point to be the point of contact. The positions of the point when the centre is at C_2, C_3 - - - are the points of intersection of the circles drawn from C_2, C_3 - - - as centres, with lines parallel to A B, drawn through the points 2, 3 - - - respectively.

Epi- and Hypo-cycloids. When a circle rolls upon the circumference of a fixed circle, the path of a point on the circumference of the rolling circle is called an Epi- or a Hypo-cycloid, according as the circle rolls upon the convex or concave side of the fixed one.

Figs. 2 and 3 show an Epi- and a Hypo-cycloid, the rolling circle and the fixed circle being 1″ and 3″ diameter respectively. Let O be the centre of the fixed circle, and B_5 and C_5 the centres of the moving circles in their middle positions. The curves subtend an angle of $\theta°$ at the centre O, which in this case is 120°.

For $O A \times \theta° = A B_5 \times 360° = A C_5 \times 360°. \quad \therefore \theta° = \dfrac{A B_5 \times 360°}{O A} = 120°$

Divide the arcs described by the centres of the rolling circles, and also the circumferences of these circles into any number of equal parts (say 8). The method of drawing the curves is shown clearly in the drawing.

When the diameter of the rolling circle is half that of the fixed circle, the hypo-cycloid is a diameter of the fixed circle.

Fig. 4.—The Involute of a circle is the curve traced out by the end of an inextensible string unwound from a circle. The string is always a tangent to the circle from which it is unwound. Suppose the curve to start at the point 1. The length of the tangent, from its point of contact, to the curve, is equal to the length of the arc of the circle from the point of contact to the starting point. The normal to the curve, at any point, is the line passing through that point and tangent to the circle.

EXERCISES.

1.—Draw the **cycloid** when the rolling circle is $1\frac{3}{4}$″ diameter.

2.—Draw the **Epi- and Hypo-cycloids,** when the diameters of the rolling and fixed circles are $1\frac{1}{2}$″ and 4″ respectively. Show the specia case of the Hypo-cycloid, when it is a straight line.

3.—Draw the **involute** of a circle of $1\frac{1}{2}$″ diameter for $\frac{3}{4}$ of a revolution.

WHEEL GEARING.

Toothed wheels are employed to communicate motion, from one shaft to another, by direct contact. **Spur wheels** connect parallel shafts; and **bevel wheels** connect any two shafts whose axes intersect.

When two spur wheels, with teeth accurately shaped, are in gear, the angular velocity ratio is constant, and is the same as if they were friction wheels having the same centres. The surfaces of the imaginary friction wheels are called the **pitch surfaces** of the spur wheels; and a plane, at right angles to the axes of rotation, intersects these surfaces in circles, which are the **pitch circles**. The **pitch** of the teeth is the distance from the centre of one tooth to the centre of the next tooth, measured along the pitch circle. The diameter of the pitch circle, in terms of the pitch and the number of teeth, is expressed by

$$d = \frac{nP}{\pi} \quad \text{where} \quad \begin{aligned} d &= \text{diameter of circle in inches.} \\ P &= \text{pitch of teeth} \quad ,, \\ n &= \text{number of teeth on wheel.} \end{aligned}$$

The dimensions of the teeth, in terms, of their pitch, are given in **Fig. 5**, and are such that there is always contact between two or more teeth on each wheel in gear.

Fig. 6 shows a scale for determining the proportions for various pitches. The difference between the tooth thickness and the tooth space, measured on the pitch line, is the *side clearance*. The acting surface of a tooth, projecting beyond the pitch circle, is called its *face*; that inside the pitch circle, its *flank*. In all the examples given, the exact form of the tooth is determined; and the teeth afterwards set out by means of circular arcs, which are approximately coincident with the correct curves. In a drawing, where a wheel occurs to a small scale, the teeth may be drawn as shown in Fig. 5.

In order that wheels may work correctly together, and have a constant velocity ratio, it is necessary and sufficient that the common normal to two teeth, at the point of contact, always passes through the point of contact of the pitch circles. This condition is satisfied if the faces and flanks of the teeth are shaped to cycloidal or involute curves.

Cycloidal Teeth.—Referring to Fig. 8, let A and B be the centres of two spur wheels, the pitch circles touching at the point P; C, the centre of a third circle, touching the other two at P. Suppose one of these circles to revolve round its centre, and cause the other two to turn round their centres in the directions shown.

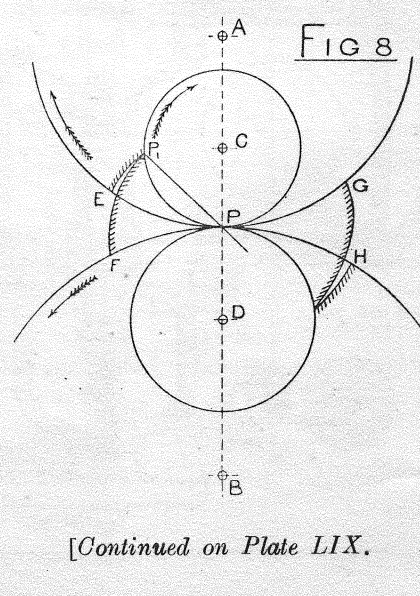


FIG 8

[*Continued on Plate LIX.*

FORM & PROPORTIONS OF TEETH FOR WHEEL GEARING.

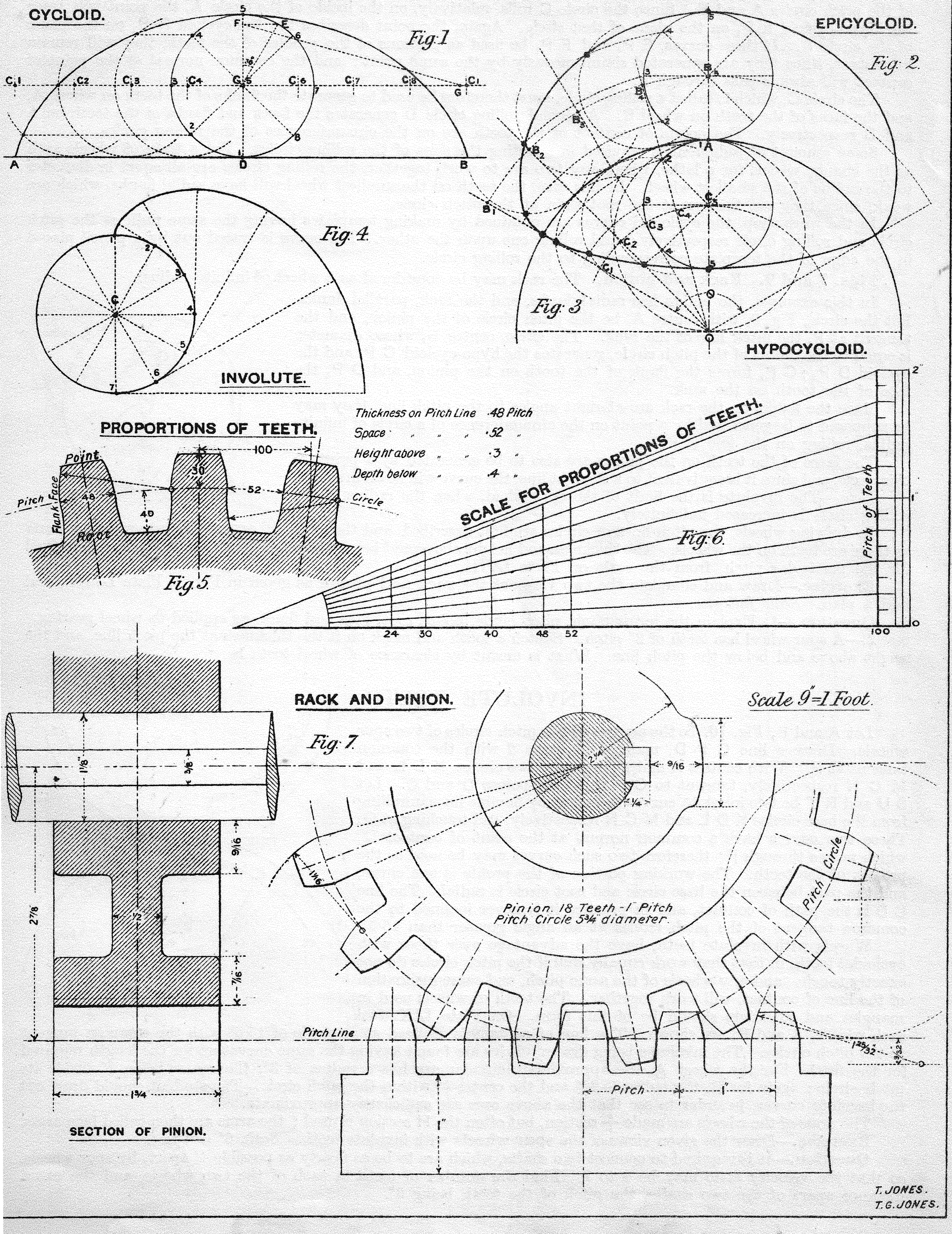


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Plate LIX.—WHEEL GEARING—*Continued.*

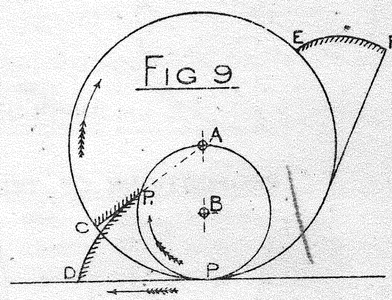
Take any point on the circumference of the circle C, and consider the curves traced out by it on the planes of the pitch circles A and B. Since the circle C rolls, relatively, on the inside of the circle A, the point will trace the Hypo-cycloid E P_1 on the plane of that circle. Again, the point describes the Epi-cycloid F P_1 on the plane of the circle B. If these curves, E P_1 and F P_1 be used as portions of the profiles of the teeth, they will remain in contact, since they are generated simultaneously by the same point; and the common normal at the point of contact will always pass through P.

The circle C, which is called a rolling circle, must therefore be used to generate the flanks of the teeth on wheel A, and the faces of the teeth on wheel B. A second rolling circle D generates the faces and flanks of the teeth on A and B respectively. The points of contact of the teeth are on the circumferences of the rolling circles.

Some amount of judgment is required in selecting the sizes of the rolling circles. For a train of wheels such as the change wheels for a lathe, where any two are to work together, the rolling circles are all equal in diameter to the radius of the smallest wheel. In this case, the teeth on the smallest wheel will have radial flanks, which are weak, since they are thinner at the root than at the pitch circle.

In the work-shop, the cycloidal curves are obtained by making templates having the same radii as the pitch circle and rolling circle respectively; and rolling one upon the other. The curve is traced out by a pencil placed in the edge of the template corresponding to the rolling circle.

Figs. 7 and 9. Rack and pinion. The rack may be considered as a wheel of infinite radius.



In this example, the pinion has radial flanks, and the rack, parallel flanks. Let the circle, Fig. 9, with centre A, be the pitch circle of the pinion, and the tangent at P, the pitch line of the rack. The circle, centre B, whose diameter is equal to the radius of the pitch circle, generates the hypo-cycloid C P_1 and the cycloid D P_1; C P_1 forms the flank of the tooth on the pinion, and D P_1 the face of the tooth on the rack.

Since the flanks on the rack are at right angles to the pitch line, they may be supposed to be generated by a point on the circumference of a circle of infinite radius, rolling on the line D P.

The faces of the teeth on the pinion are also to be generated by a point on this circle, and since it is equivalent to a straight line, the curve will be an involute.

Note.—The involute is the limit of the epi-cycloid, when the radius of the rolling circle is increased indefinitely.

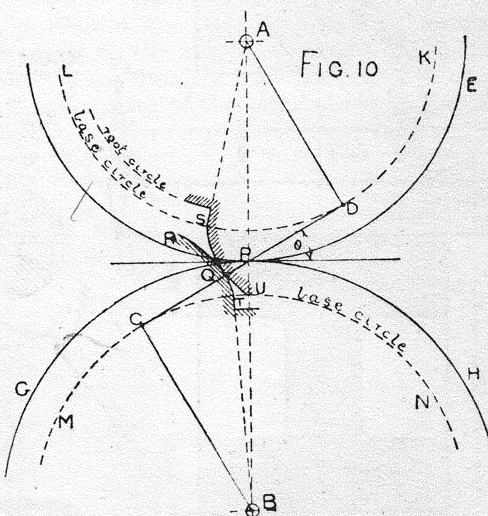
In drawing wheels, the pitch lines or circles are first described, and then divided accurately into as many parts as there are teeth on the wheels. The thickness and height above and below the pitch line of the teeth are determined for the particular pitch, from the scale on Plate LVIII.

Exercise.—Draw and complete the two views of the **rack and pinion** as given in Fig. 7, Plate LVIII., and add a plan. *Scale full size.*

Question.—1. Explain the terms pitch, pitch circle, point, root, face and flank, as applied to wheel gearing.

2.—A spur wheel has teeth of 2″ pitch. Sketch a tooth and mark on it the thickness at the pitch line, and the height above and below the pitch line. What is meant by clearance of wheel teeth?

INVOLUTE TEETH.



Let A and B, Fig. 10, be the centres of the pitch circles of two spur wheels. Draw a line C P D, making an angle θ with the common tangent at P. With centres A and B, draw the two circles K D L and M C N respectively, tangent to C P D, at the points D and C. Let S U and R T be two involute curves described by flexible lines unrolling from the base circles K D L and M C N respectively, and touching at Q. These two curves have a common normal at the point of contact Q, which passes through P; therefore two such curves may be used as the profiles of the teeth. The working portion of the profile is one curve, and the part between the base circle and root circle is radial. The line C D is the path of contact, and in practice it is never inclined to the common tangent of the pitch circles at an angle greater than 15·5°.

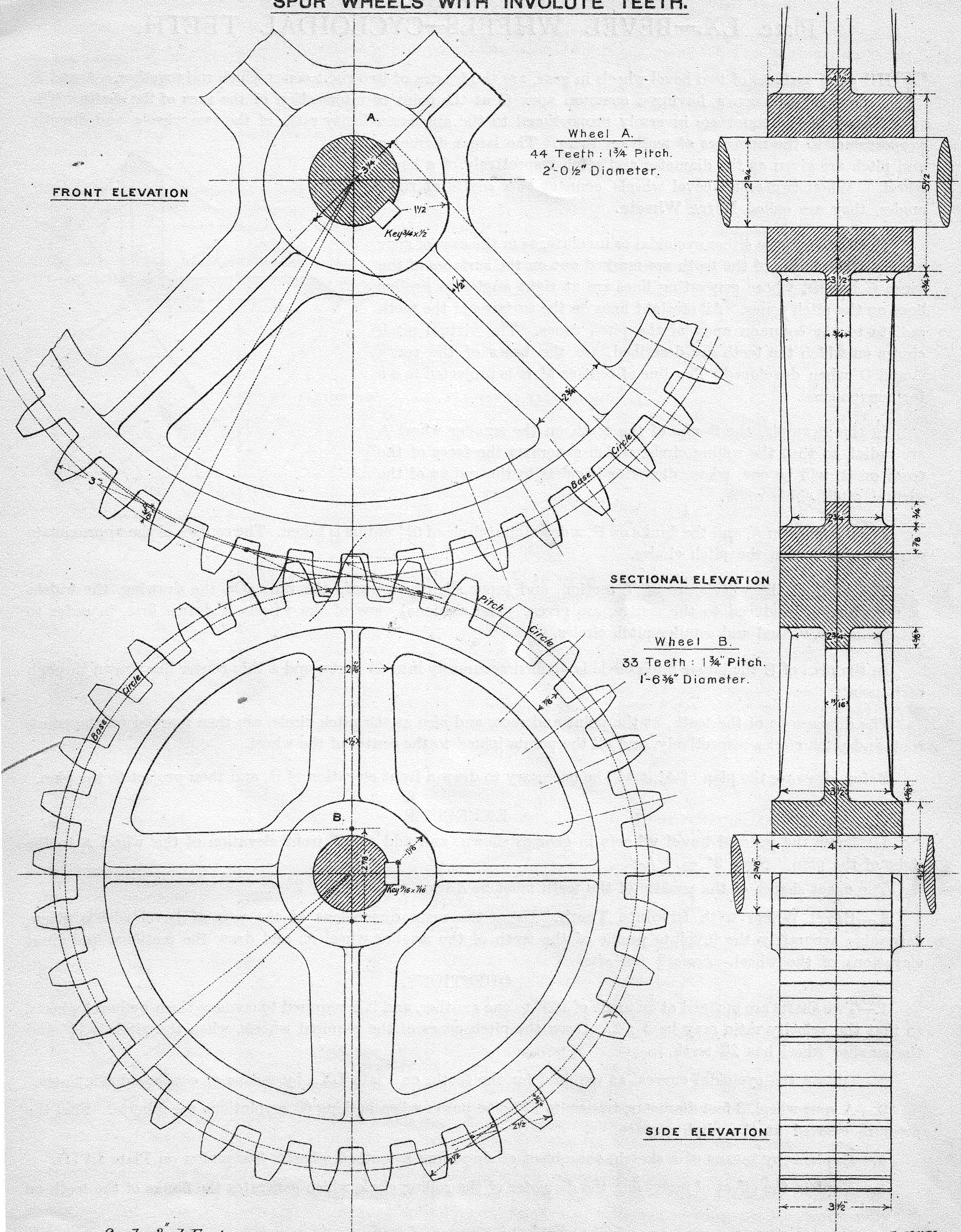
Wheels with involute teeth, have the advantage over those with cycloidal teeth, in that they work equally well if the pitch circles do not exactly touch; and any wheels of the same pitch, and same inclination of the line of contact, will work together. The teeth of wheels used in mangles and calenders should be of this form. On Plate LIX. two such wheels, A and B, are shown. The line of contact is inclined at an angle of 15·5° with the common tangent to the pitch circles. The involutes being drawn, circles are found having the same curvature for the length required for the teeth. For the wheel A, the approximate circular arc has a radius of 3″, the centre being $\frac{3}{8}$″ inside its pitch circle; again for B, the radius is $2\frac{1}{2}$″ and the centre $\frac{1}{4}$″ within the pitch circle. The student should draw out the involute curves, in order to see that the above arcs are sufficiently approximate.

The arms of the wheels are made **+** section, but often the H section is used; the arms are then called face arms.

Exercise.—Draw the given views of the **spur wheels** with involute teeth. *Scale 6″ = 1 foot.*

Question.—It is required to connect two shafts, which are to be as nearly as possible 4′ apart, by spur wheels, so that the velocity ratio may be 4 to 1. Find the number of teeth in each of the two wheels, and the exact distance apart of the two shafts, the pitch of the teeth being 3″.

SPUR WHEELS WITH INVOLUTE TEETH.



Scale 3" = 1 Foot

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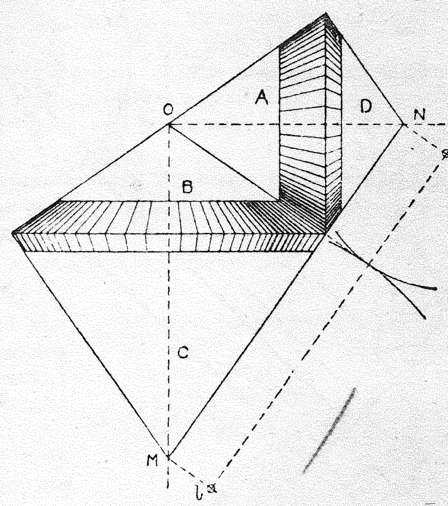
Plate LX.—BEVEL WHEELS—CYCLOIDAL TEETH.

THE pitch surfaces of two bevel wheels in gear, are the frustra of two right cones which roll together. A and B represent two cones, having a common apex O at the point of intersection of the axes of the shafts. The cases of the cones are inversely proportional to the angular velocity ratio of the two wheels, and directly proportional to the numbers of teeth on them. The larger diameter and pitch are given as the diameter and pitch respectively of a bevel wheel. When two equal bevel wheels connect two shafts at right angles, they are called **Mitre Wheels.**



The teeth may be either cycloidal or involute, as in the case of spur wheels. The forms of the teeth are marked out on the surfaces of the cones C and D, whose generating lines are at right angles to similar lines on the pitch cones. All straight lines on the surfaces of the teeth radiate to the common apex of the pitch cones. The virtual pitch circles on which the teeth are described, are the bases of the cones C and D when developed. The line of centres M N is projected to *a b* for convenience.

In this example, the flanks of the teeth on the smaller wheel A are radial, so that the rolling circle, which generates the faces of the teeth on B, will be one, whose diameter is equal to the radius of the virtual pitch circle of A.

For the faces on A, and the flanks on B, a describing circle of $5\frac{1}{2}''$ radius is taken. The centres of the approximate circular arcs are on the pitch circles.

The arms of the wheels are of T section, and taper slightly towards the rim. On the drawing, the widths of the arms, if produced to the centre, are given. The sectional view of the wheels is drawn first, in order to determine the virtual and smaller pitch circles.

In the plan of B the larger pitch circle is divided accurately into 45 parts, and a thin centre line drawn through each point.

The dimensions of the teeth, at the point and root, and also at the pitch circle, are then marked on the point, root, and pitch circles respectively, and all the points joined to the centre of the wheel.

Before drawing the plan of A, it will be necessary to draw a front elevation of it, and then project to the plan.

EXERCISES.

1. Draw the **pair of bevel wheels** in gear as shown, and add a half front elevation of the wheel A to the right of the plan. *Scale* $3'' = 1$ *foot*

The exact shapes of the profiles of the teeth must be drawn.

2.—**Bevel Wheel with Involute Teeth.** Using the main dimensions of the pair of bevel wheels given, determine accurately the involute profile of the teeth of the smaller wheel A, and draw the sectional and front elevations of the wheel. *Scale* $\frac{1}{2}$ *full size.*

QUESTIONS.

1.—Two shafts are inclined at an angle of 120° to one another, and it is required to connect them by bevel wheels, so that the velocity ratio may be 3 : 2. Draw the pitch cones of the required wheels, when the pitch is $1\frac{1}{2}''$ and the smaller wheel has 26 teeth.

2.—Obtain the cycloidal curves, as required for the teeth, on Plate LX., by means of cardboard templates.

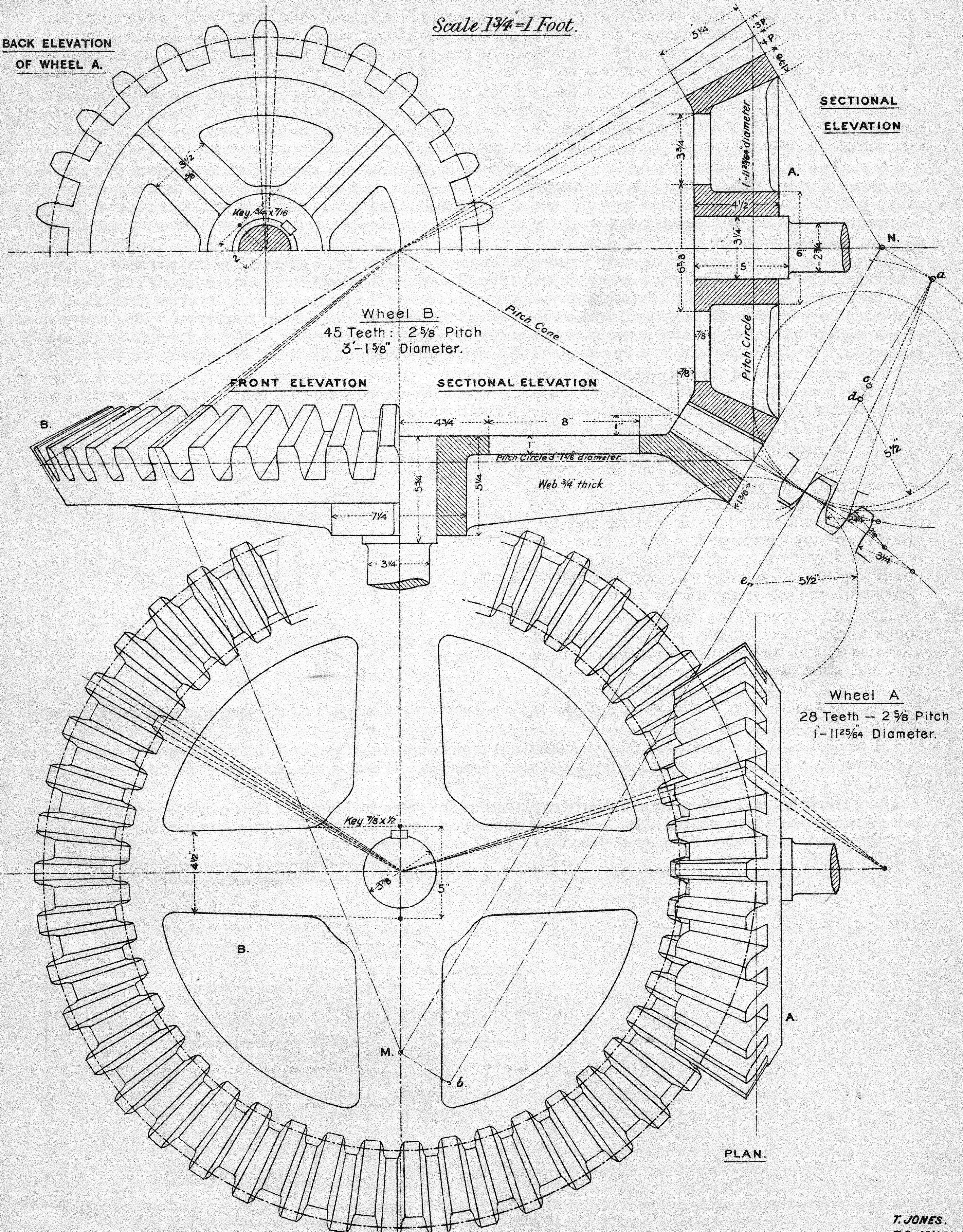
3.—A spur wheel, 3 feet diameter, transmits 15 horse power when making 60 revolutions per minute. Find the pressure exerted at the pitch surface.

4.—Explain, by means of a sketch, the object of shrouding a toothed wheel. See pinion on Plate LVIII.

5.—Explain the effect of increasing the diameter of the rolling circle which generates the flanks of the teeth on a wheel.

BEVEL WHEELS WITH CYCLOIDAL TEETH.

Scale $1\frac{3}{4}$" = 1 Foot.



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Plates LXI., LXII., LXIII. and LXIV.—FREEHAND SKETCHING.

THE ability to make good freehand sketches of engineering details is of great value both to the student and the professional draughtsman ; and with the view of providing the former with suitable exercises four sheets of isometric sketches are given. **These sketches are to serve the purpose of models, by reference to which the required orthographic views are to be sketched in correct projection and in fair proportion.**

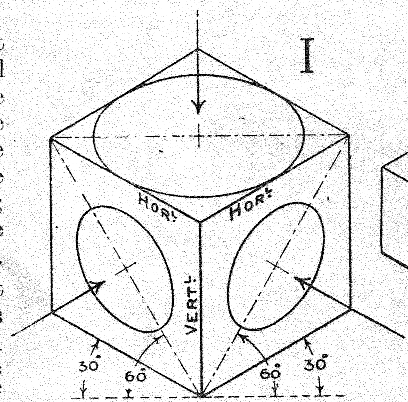
The use of models is doubtless of value to a student who is commencing the work with practically no training in Geometry or science generally. The average engineering student, however, has usually a fair knowledge of Practical Geometry, and is familiar with the details he is about to draw—from his work in the workshop—and it would then appear that the frequent resort to models is quite unnecessary, and, in fact, reflecting upon his power of imagination.

A student may be given a model and required to make dimensioned sketches of the various orthographic projections, and from the sketches prepare accurate scale drawings. But such a procedure is rarely necessary : it not only limits considerably his drawing work, and thus retards his acquisition of a neat and clear style of drawing, but makes no demand upon his imagination, and so unfits him for *reading* a new drawing or forming a mental picture of a mechanical detail he has never seen.

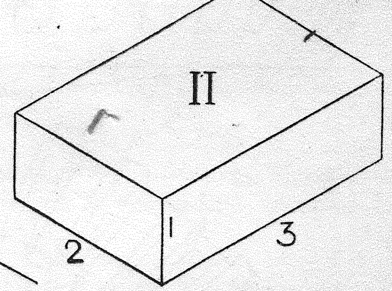
If, by a careful and systematic early training in Engineering Drawing, a student has the power of accurately interpretating a drawing, he may acquire a wide knowledge of machine construction by a careful study of well-selected scale drawings ; and this without devoting very considerable time to the making of scale drawings of all the details of which a knowledge would be valuable. Also, if a student wishes to test quickly his knowledge of the construction of any engineering detail, let him make sketches of the necessary orthographic projections ; and the degree of success with the sketching will be a fair gauge of his useful knowledge of the detail in question.

To make freehand orthographic views from carefully prepared isometric drawings makes a demand upon the imagination—without which an engineer would be useless—and to ensure that the student may judge accurately of the shapes and relative sizes of the various parts, it is necessary to explain a few simple points on the subject of Isometric Projection.

I



II

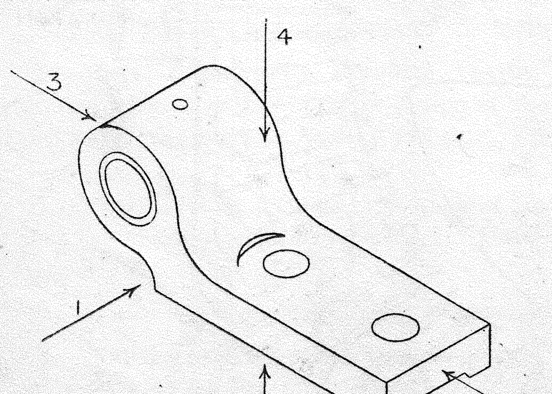


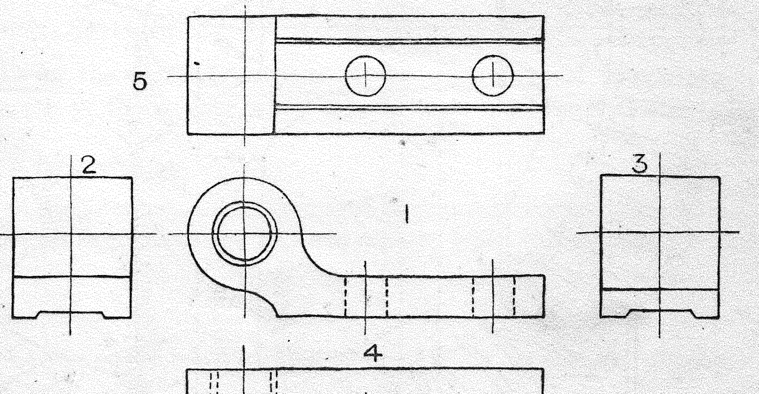
An **isometric projection** of an object is a view from such a direction that three equal lines mutually at right angles project into three equal lines equally inclined to one another. One of the three reference lines is vertical and the other two are horizontal. Such lines are represented by the three adjacent edges of a cube ; and if the cube be standing on a horizontal plane its isometric projection would be as given in Fig. 1.

The directions of the arrows are at right angles to the three mutually perpendicular faces of the cube, and indicate the directions in which the solid must be viewed for the orthographic projections. If in the given isometric drawing of a rectangular solid—Fig. 2—the lengths of the three adjacent edges are as 1 : 2 : 3, then the solid would have :—height : width : length = 1 : 2 : 3.

A circle drawn on a horizontal face of a solid will project into an ellipse, with its major axis horizontal ; and one drawn on a vertical face will also project into an ellipse with its major axis inclined 60° to the horizontal—see Fig. 1.

The Principles of Projection are clearly explained in the notes to Plate II. ; but a simple example is given below, where the views obtained by looking at the object—as represented by the isometric drawing—in the horizontal and vertical directions are sketched, to a small size, in correct projection.



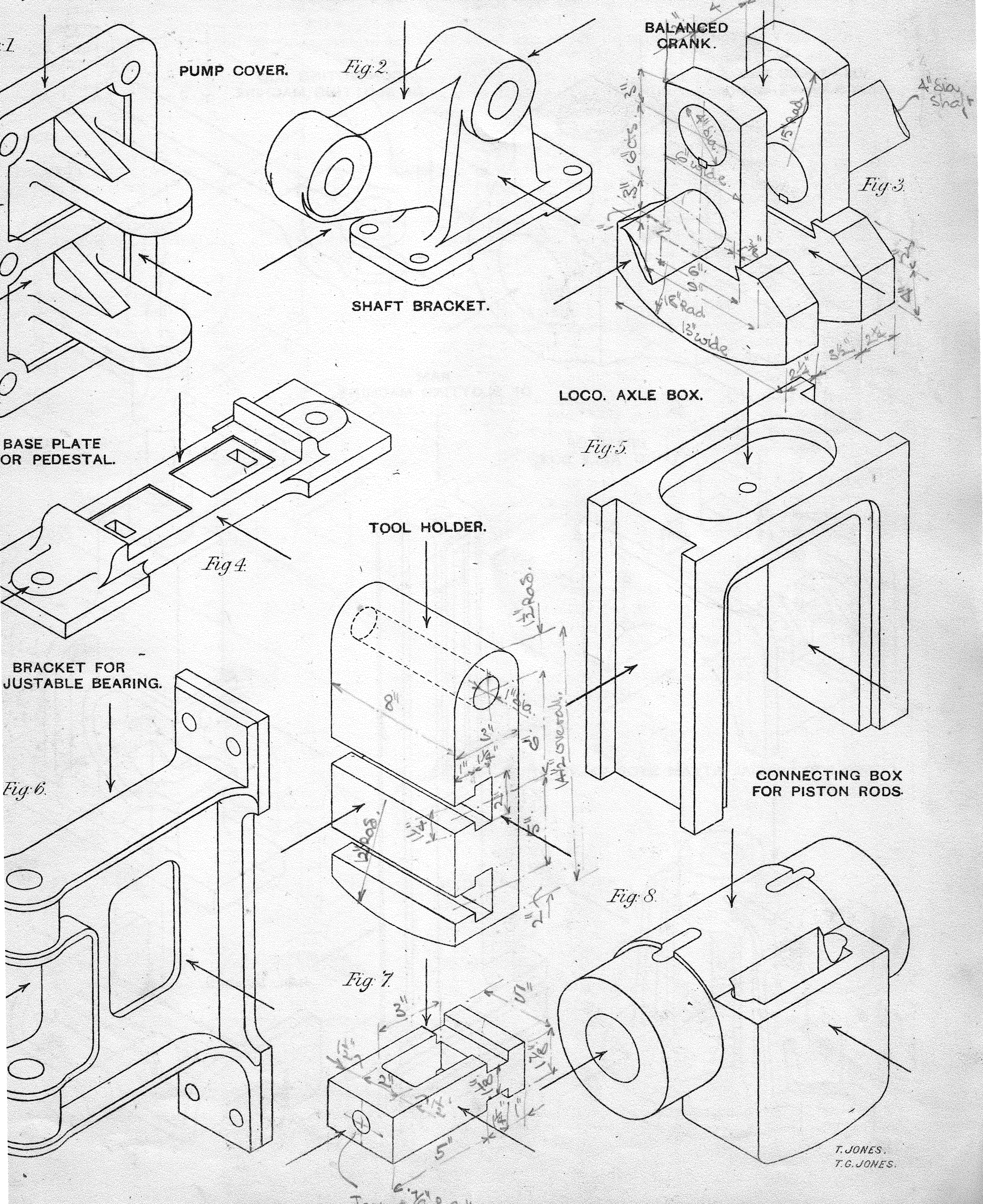


For each of the examples, given on Plates LXI., LXII., LXIII., LXIV., sketch the views required—as for the above example—and note the materials of which the parts would most probably be made.

EXAMPLES FOR FREEHAND SKETCHING.

FOR GENERAL INSTRUCTIONS SEE OPPOSITE PAGE LXI.

N.B.—On no account must the Isometric Views given below be copied; they are to serve the purpose of Models from which the various Orthographic Projections may be obtained.



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Book I

EXAMPLES FOR FREEHAND SKETCHING.

PLATE LXII.

FOR GENERAL INSTRUCTIONS SEE PAGE FACING PLATE LXI.

N.B.—On no account must the Isometric Views given below be copied; they are to serve the purpose of Models from which the various Orthographic Projections may be obtained.

VALVE ROD END
(JOY'S VALVE GEAR).

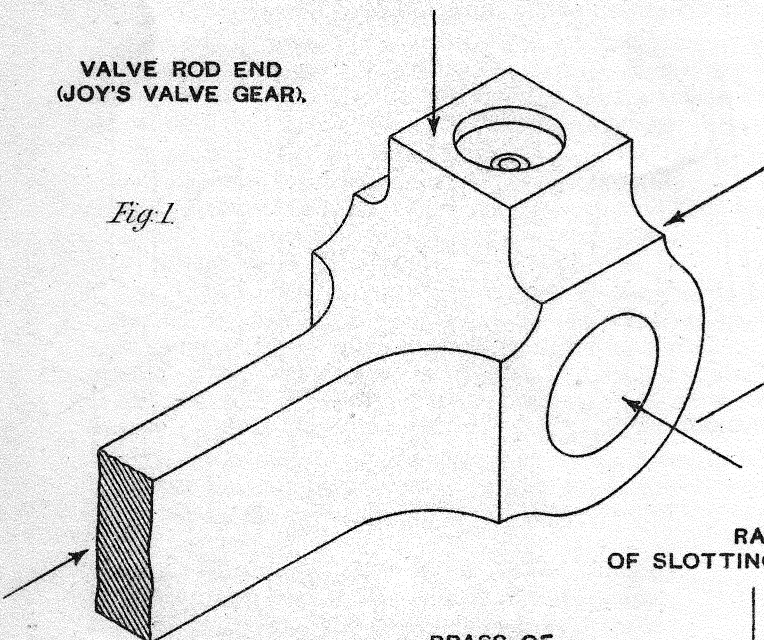

Fig: 1.

CONNECTING ROD
OF SLOTTING MACHINE.

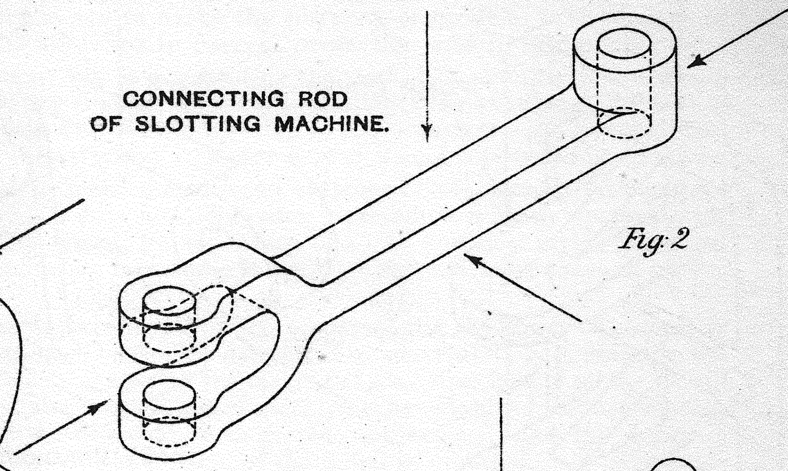

Fig: 2.

BRASS OF
LOCO AXLE BOX.

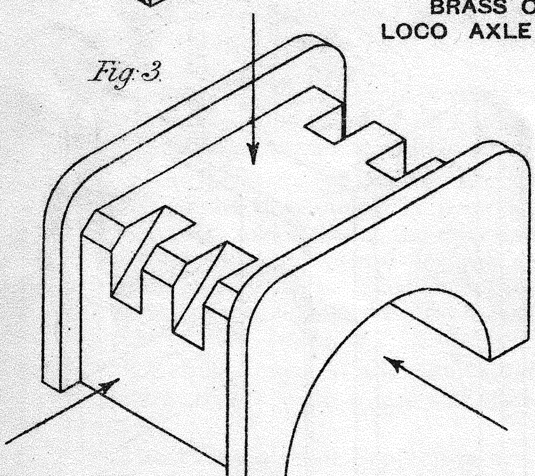

Fig: 3.

RAM
OF SLOTTING MACHINE.

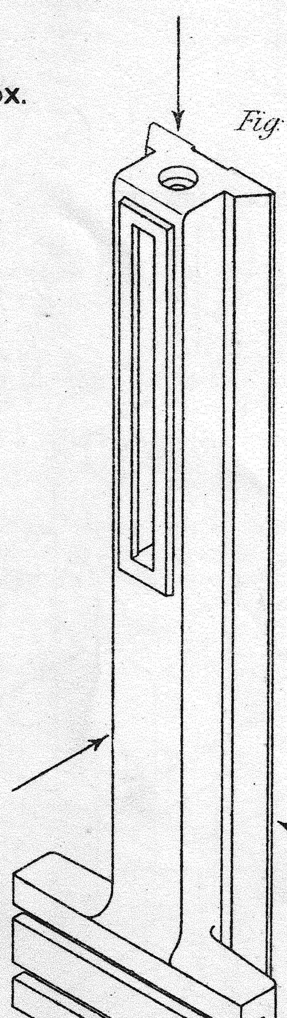

Fig: 4.

PIPE VICE.

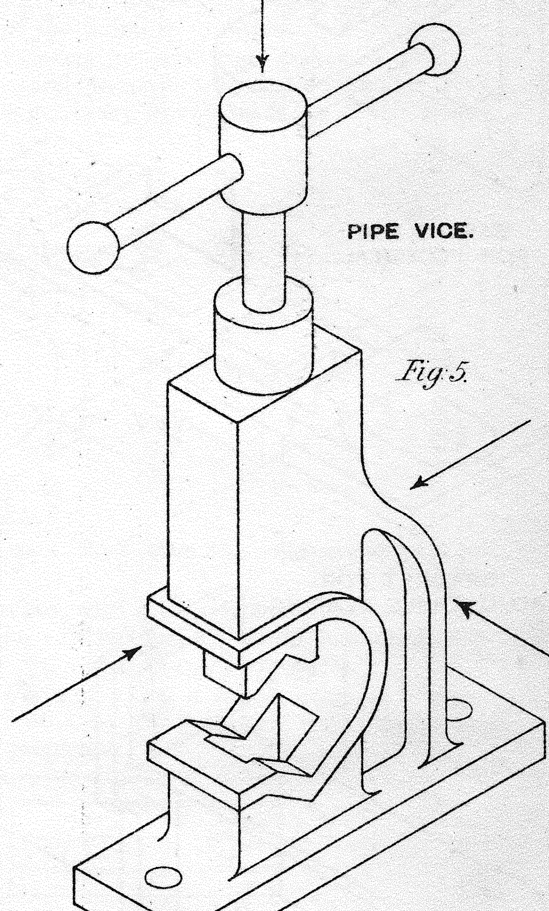

Fig: 5.

COVER FOR SPECIAL STEAM STOP VALVE.

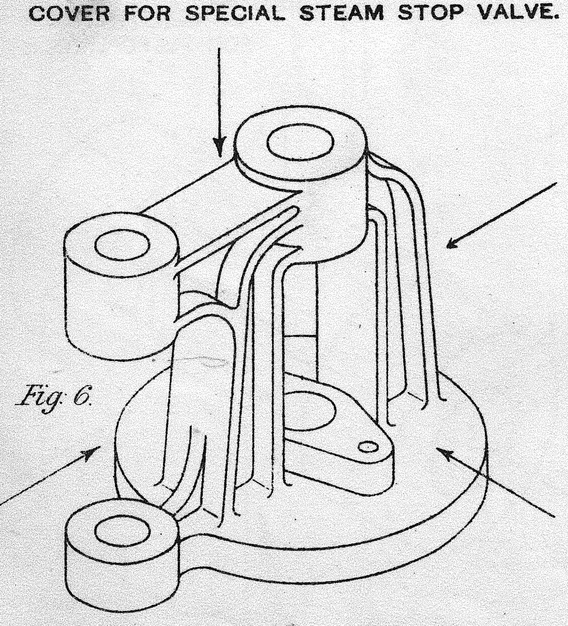

Fig: 6.

RAIL BENDER.

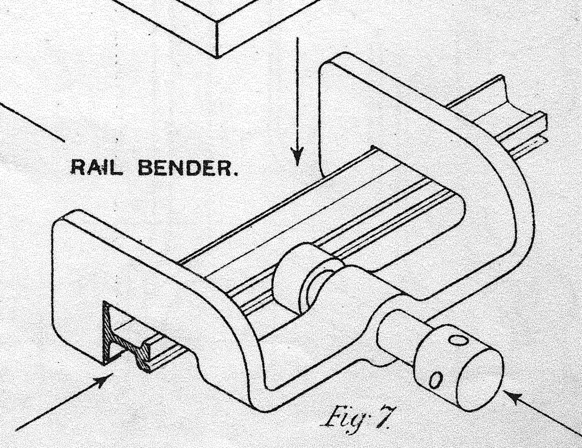

Fig: 7.

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EXAMPLES FOR FREEHAND SKETCHING.

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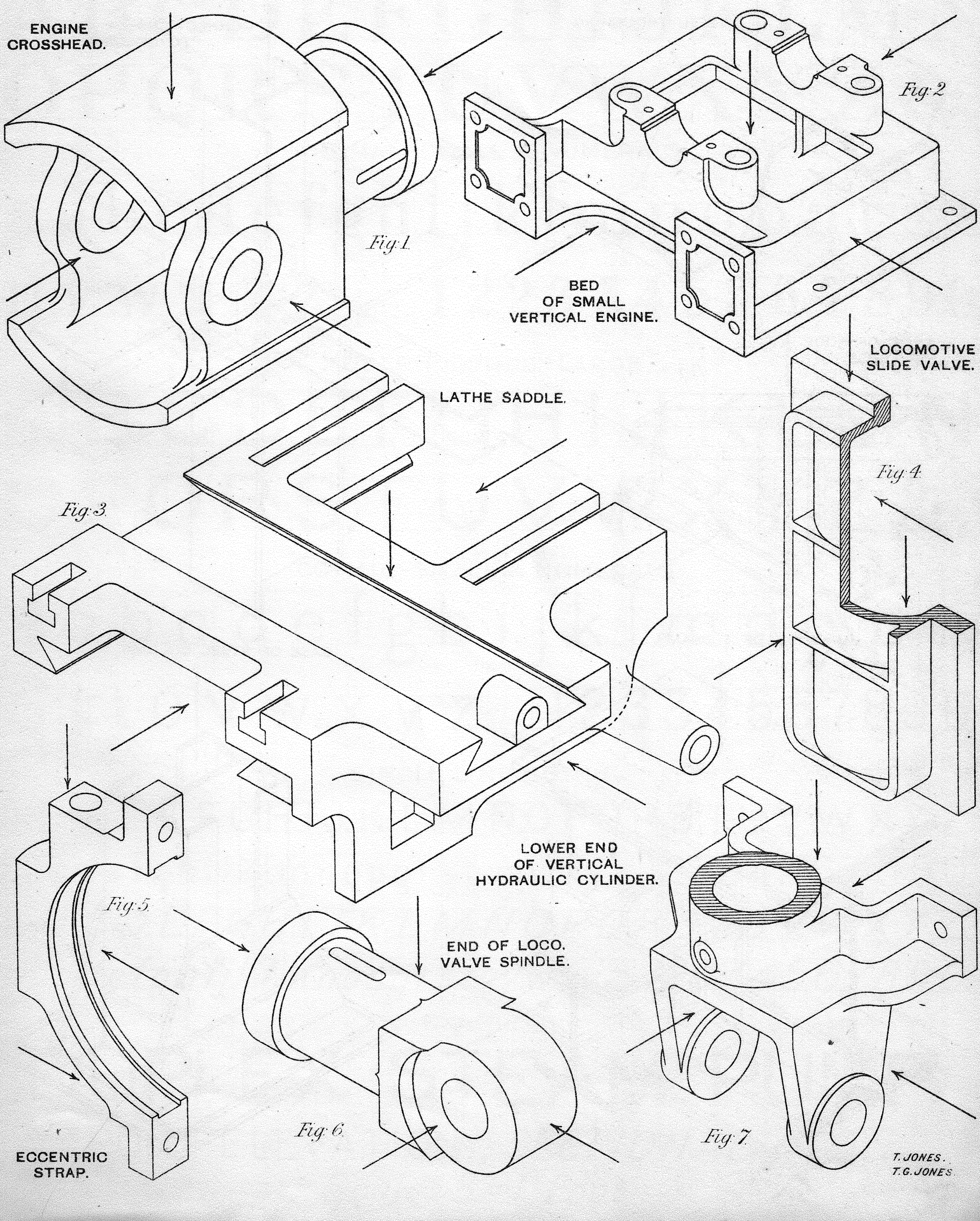
N.B.—On no account must the Isometric Views given below be copied; they are to serve the purpose of Models from which the various Orthographic Projections may be obtained.



EXAMPLES FOR FREEHAND SKETCHING.

FOR GENERAL INSTRUCTIONS SEE PAGE FACING PLATE LXI.

N.B.—On no account must the Isometric Views given below be copied; they are to serve the purpose of Models from which the various Orthographic Projections may be obtained.

STEEL TROUGHING

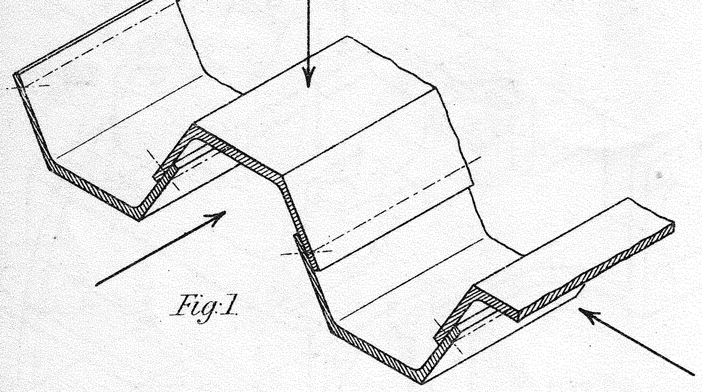


Fig:1.

CONNECTIONS OF I BEAMS TO COLUMN.

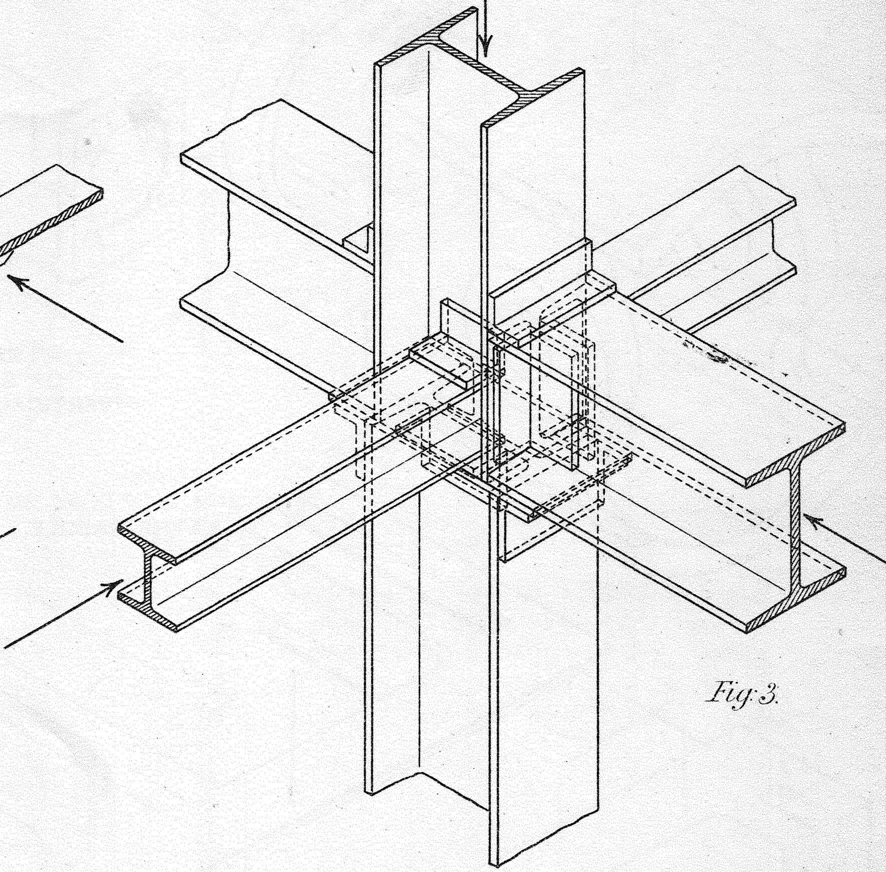
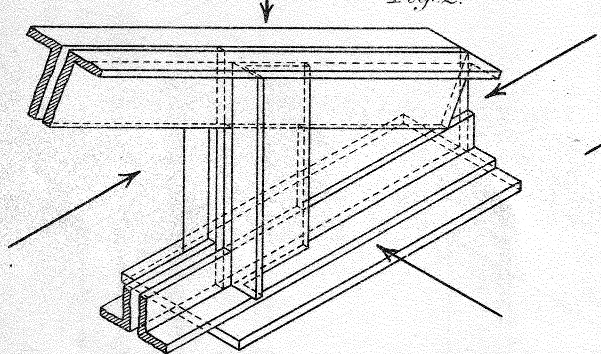


Fig:3.

SHOE OF ROOF TRUSS.

Fig:2.



JUNCTION OF I BEAMS.

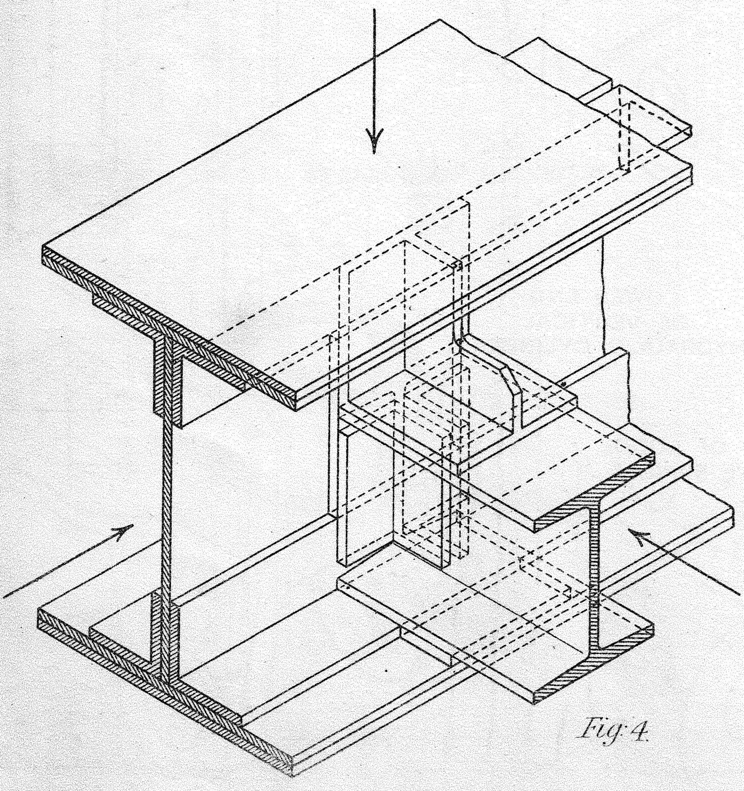


Fig:4.

BASE OF STEEL COLUMN.

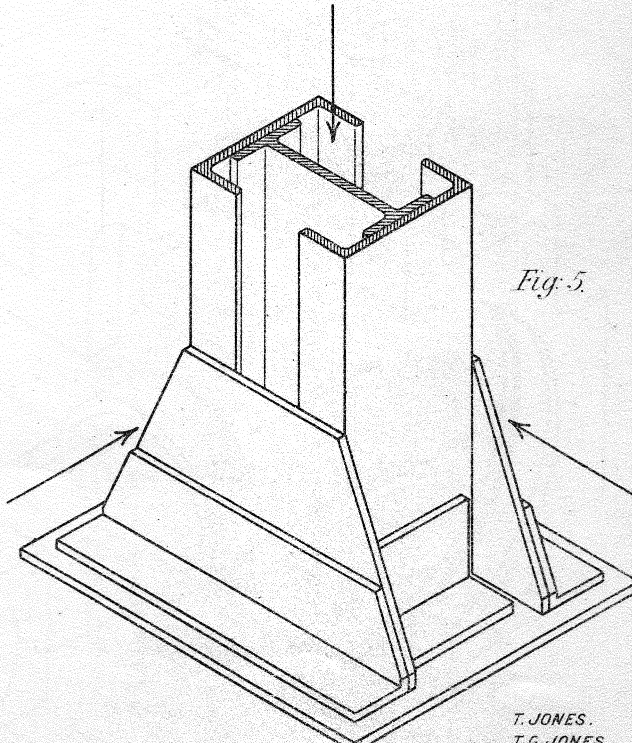


Fig:5.

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FREE-HAND LETTERING SUITABLE FOR ENGINEERING DRAWINGS.

MODERN ROMAN-LARGE.

ABCDEFGHIJKLMN
OPQRSTUVWXYZ&.

ROMAN-SMALL & NUMERALS.

aabcdefghijklmnopqrst
uvwxyz. 12233455567890

MODERN GOTHIC-LARGE.

ABCDEFGHIJKLMN
OPQRSTUVWXYZ&.

GOTHIC-SMALL & NUMERALS.

aabcdefghijklmnopq
rstuvwxyz. 12334567890

SINGLE LINE GOTHIC.

ABCDEFGHIJKLMNOPQRSTUVWXYZ.
abcdefghijklmnopqrstuvwxyz. 12334567890.
ABCDEFGHIJKLMNOPQRSTUVWXYZ.
abcdefghijklmnopqrstuvwxyz. 12334567890.

EXAMPLES.

DUPLEX STEAM PUMP.

FRONT ELEVATION. *SECTIONAL PLAN.*

Scale 3"=1 Foot.

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Plates LXVIII. and LXIX.—REPRESENTATION OF VARIOUS MATERIALS AND SURFACES.

THE work of an engineer's drawing office, apart from that of calculation, consists in the preparation of pencil and ink drawings and tracings ; the last-named being either for use in the shops, or for the purpose of producing duplicate blue print copies by a simple photographic process. Tracings or blue prints made from them are also used for sending out with machines as a guide to their erection and placing in the mill or workshop.

Ink drawings and tracings are almost invariably coloured, the sectional parts being covered with flat washes of those colours, which custom has decided shall be used to represent the various materials, and the flat and curved surfaces tinted so that one may, from the drawing, form a good idea of the shapes of the parts.

In the preparation of a standard drawing, the views of the curved surfaces are carefully shaded with thin washes of indian ink, ground down from the stick, and afterwards, all portions of the drawing are tinted with the distinctive colours. A little judicious shading for the representation of shadows renders the drawing very effective.

After the completion of the colouring, the centre lines are drawn in red colour (crimson lake or scarlet), and the dimension lines in Prussian blue. (*See coloured drawing of hydraulic cylinder and ram.*)

On the accompanying plate, Fig. 1, are given the colours universally adopted for the representation of the materials used in engineering work. Distinction between the different materials is rendered more evident by the character of the section lines ; the coarser the material, and the larger the area to be sectioned, the greater the pitch of the section lines Sometimes on shop drawings and tracings the section lines are drawn in colour with a brush, or else omitted ; the latter system, however, often leads to a misunderstanding concerning the limitations of the various parts.

For the colouring of small drawings it will be necessary to have at least three sizes of brushes ; Nos. 10, 8, and 6 are perhaps the most useful. For large flat washes the largest size of brush will give the best results, and when moistened with clear water, will serve for " softening off " the large shaded surfaces.

Figs. 2 to 9 inclusive show how the exact forms of the various geometrical surfaces are suggested by shading.

The parallel rays of light illuminating the surfaces are assumed to pass over the left shoulder of the observer, and have such an inclination that their projections make 45° with both plan and elevation (*see notes on shade lining*), but to avoid confusion, and without detracting from the appearance of the drawing, it is usual to shade all views alike, assuming the light to proceed from the left-hand top corner of the drawing. If it be remembered that the intensity of light on any portion of a surface is greater the greater the angle between it and the rays, there will be no difficulty in understanding the shading of most curved surfaces. The lightest and darkest portions will therefore be those which are respectively at right angles to, and parallel to the direction of the light.

From the lightest to the darkest portion the colour will gradually increase in intensity—the gradation of colour being produced by the process of " softening off." In the shading of any curved surface, when the positions and shapes of the lightest and darkest portions have been determined, a flat wash of colour is put on the figure, commencing at the darkest part and proceeding towards the lightest ; and when a little distance from the latter part the wet edge of the coloured is softened off. Succeeding flat washes are treated in the same manner, but each finishes further from the lightest portion than the preceding one.

If the drawing is to be highly finished, the effect of pure colours must be lessened by first shading parts with thin washes of colour made by rubbing a stick of indian ink.

If the surface to be shaded is large, a good effect is produced by using flat washes without softening off, each wash being narrower and nearer the darkest region than the preceding one. The cylinder in Fig. 2 is shaded in this way.

In reality an object is illuminated by the light proceeding immediately from the source, and also by the light reflected from surrounding objects, so that there may be many regions of light and shade. However, those caused by reflected light would be subsidiary to those which are alone considered in the accompanying examples.

HYDRAULIC CYLINDER AND RAM.*

This particular example has been primarily chosen for the coloured plate on account of its showing clearly the application of the principles and methods mentioned above.

The drawing represents in sectional and end elevations and plan a cast-iron hydraulic cylinder and covers fitted with a ram. The cylinder, made of substantial thickness to resist the great pressure exerted by the water, is accurately bored, and the front and back covers secured to its flanged ends by steel studs. The front cover permits of the passage of the ram through it. The piston head of the ram is fitted with two " hat " leathers, and to prevent leakage past the front cover a cup leather packing ring K, is used.

Hydraulic leather packings differ from those used in the steam engine, since the pressure which the former exert to prevent the escape of the working fluid varies with the pressure of the fluid.

The cylinder and ram may be used in an hydraulic machine of two powers. The ram would exert the greater force when the water under pressure is admitted to the cylinder only through the back cover, the other end of the cylinder being connected with the waste pipe by the small passage in the cylinder barrel. If pressure be admitted to the two ends of the cylinder, the effective pressure will be that due to the difference of areas of the two sides of the piston.

EXERCISE.

Draw, colour, and dimension the given views of the **hydraulic cylinder and ram.** *Scale* 4″ = 1 *foot.*

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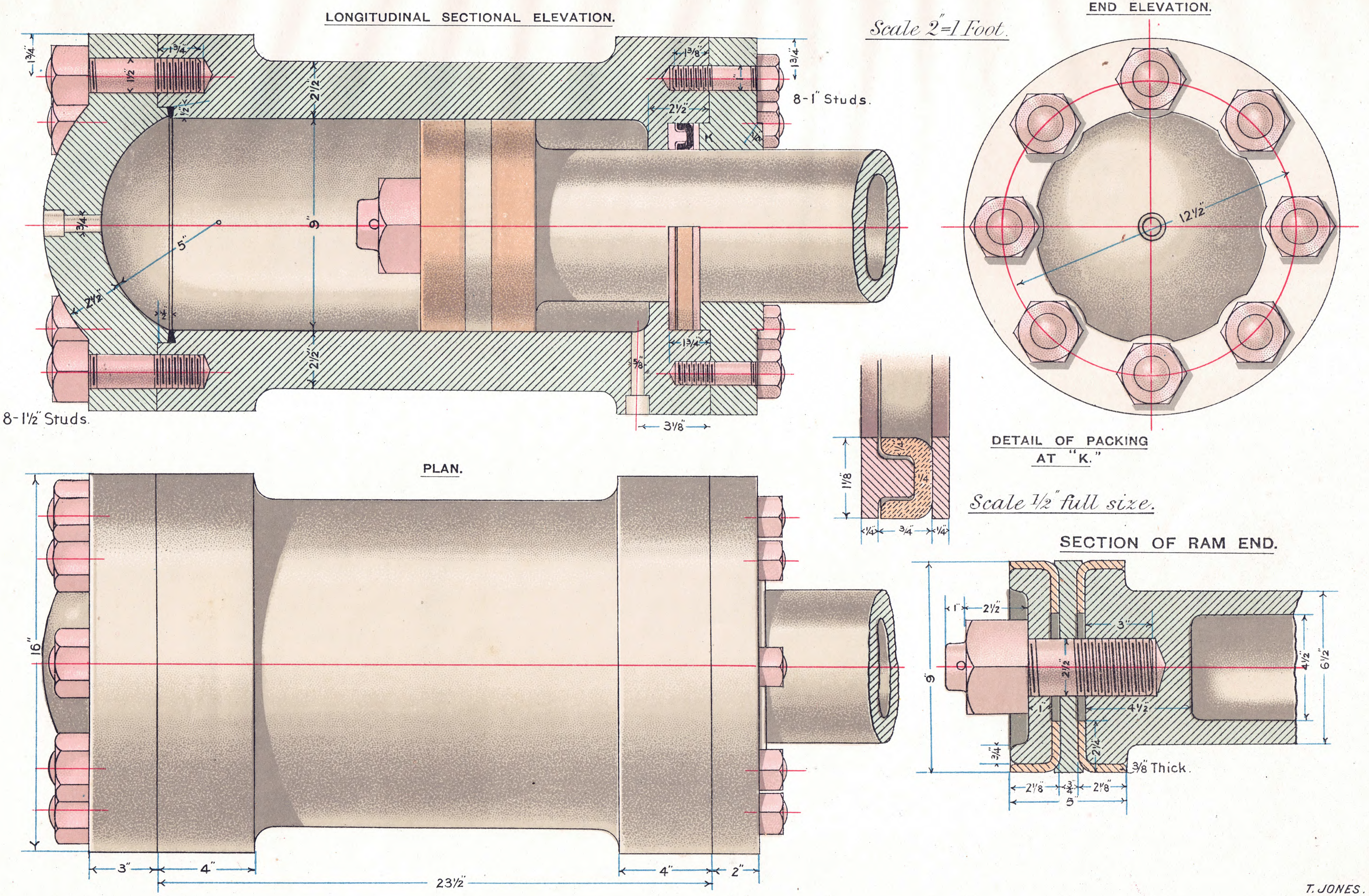
* The drawing has been prepared from one of the S. and A. examination papers in Machine Drawing by permission of the Controller of His Majesty's Stationery Office.

Book I.

PLATE LXVIII.

HYDRAULIC CYLINDER AND RAM.



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REPRESENTATION OF VARIOUS MATERIALS AND SURFACES.